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INCREASING THE VOLUMETRIC EFFICIENCY OF DIESEL ENGINES
BY INTAKE PIPES

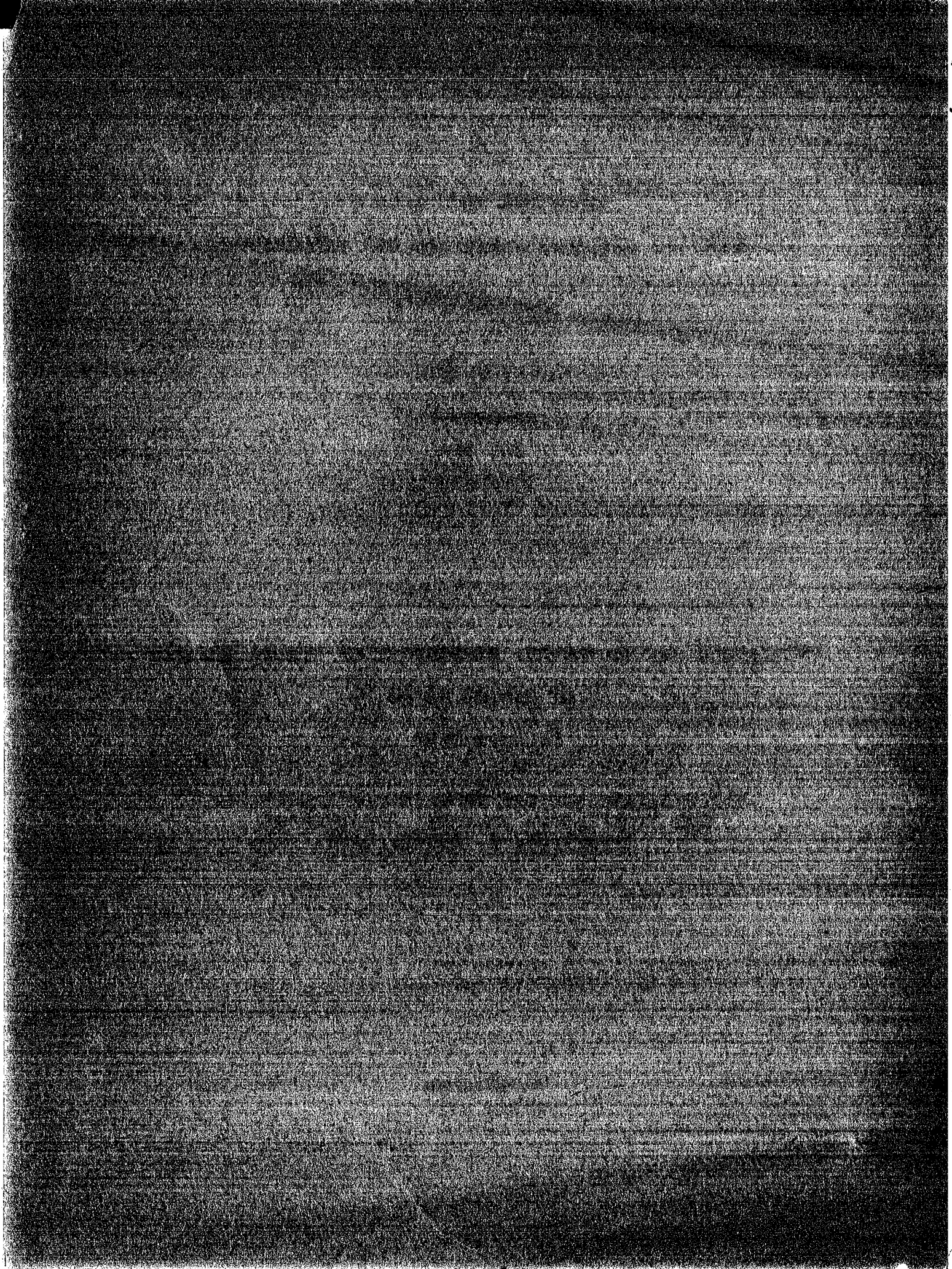
By Hans List

Mitteilungen aus den technischen Instituten
der Staatlichen Tung-Chi Universitat
Report No. 4, April, 1932

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL MEMORANDUM NO. 700

INCREASING THE VOLUMETRIC EFFICIENCY OF DIESEL ENGINES
BY INTAKE PIPES*

By Hans List

Synopsis.— Development of a method for calculating the volumetric efficiency of piston engines with intake pipes. Application of this method to the scavenging pumps of two-stroke-cycle engines with crankcase scavenging and to four-stroke-cycle engines. The utility of the method is demonstrated by volumetric-efficiency tests of the scavenging pumps of two-stroke-cycle engines with crankcase scavenging. Its practical application to the calculation of intake pipes is illustrated by an example.

INTRODUCTION

In order to utilize the piston displacement of Diesel engines better, it is sought to increase the revolution speed and the mean pressure. It is possible to increase the latter either by reducing the excess of air required for complete combustion or by increasing the air charge. Superchargers enable considerable increase in the air charge and are especially efficacious in conjunction with exhaust-gas turbines. The engines, however, are thereby made considerably more complex and expensive. With intake pipes the increase in the air charge is considerably reduced but, since the increased expense of construction is small and no additional moving parts are required, their use is often advantageous.

Intake pipes are especially advantageous for two-stroke engines in which the scavenging air is compressed by the back side of the piston. The piston displacement of the scavenging pump is then equal to that of the working cylinder, while the quantity of scavenging air is relatively small and can be increased only by improving the ef-

*"Die Erhöhung des Liefergrades durch Saugrohre bei Dieselmotoren." Mitteilungen aus den technischen Instituten der Staatlichen Tung-chi Universität, Woosung, China. Report No. 4, April, 1932.

efficiency of the scavenging pump. Any increase in the latter greatly increases the air charge and the engine power, because the increase in the air charge with respect to the scavenging air is inversely proportional to the quantity of the latter.

For a long time thorough investigations have been conducted in the engine laboratory of the Tung Chi University on the improvement of the volumetric efficiency of the scavenging pumps of engines with crankcase scavenging by intake pipes. A report of this work will appear later. Preparations were begun for conducting experiments with four-stroke engines, but had to be postponed on account of the war. Tests made by the writer about seven years ago on a 75 hp single-cylinder four-stroke-cycle engine in Europe showed an increase of about 15 per cent in the air charge with an intake pipe.

Many of the calculations and tests were made by my coworker Hsueh Dsai Hsian and some of the calculations were made by Chang I Lu.

THEORY OF THE INTAKE PROCESS

In the intake through a pipe, two vibrational systems are superposed. System I consists of the air in the intake pipe and the cylinder contents which have a shock-absorbing effect, while system II consists of the mass and elasticity of the air in the intake pipe alone. Both systems vibrate during the intake, but only system II between the intake strokes. System I has the greater effect on the intake process.

In the intake pipe (fig. 1) the fluid flows from I to II in the time dt , its velocity being affected by pressure and friction. With the density ρ and the velocity w , we have

$$-\frac{\partial p}{\partial y} w dt = \rho w \frac{\partial w}{\partial y} w dt + \rho \frac{\partial w}{\partial t} w dt + \rho w^2 r dt$$

the last term representing the friction. We finally obtain

$$-\frac{\partial p}{\partial y} = \rho w \frac{\partial w}{\partial y} + \rho \frac{\partial w}{\partial t} + \rho w^2 r$$

Aside from this equation, the following continuity equation for compressible fluids must also be satisfied.

$$\frac{\partial \rho}{\partial t} + \rho \frac{\partial w}{\partial y} + w \frac{\partial \rho}{\partial y} = 0,$$

i.e., the difference in a unit volume of the inflowing and outflowing fluid must equal the change in density of this unit volume.

There are also equations which represent the processes when the air enters the pipe and the cylinder. If p_a and w_a are the pressure and velocity at the beginning of the pipe and p_o is the outside pressure, we have

$$\frac{w_a^2}{2g} = - \int_{p_a}^{p_o} v \, dp.$$

In the cylinder the volume and quantity of the gas are affected by the motion of the piston and also by the inflow of air from the intake pipe. The corresponding piston stroke is z and the piston displacement is V_h . The volumetric change is therefore

$$dV_1 = V_h \, dz$$

and the volume of the inflowing air is

$$dV_2' = f w_i \, dt$$

in which f is the pipe section, the subscript i indicating the condition at the opening of the intake pipe into the cylinder. The kinetic energy of the inflowing air is converted into heat by eddying and the air is also heated by contact with the walls. Hence

$$dV_2 = \tau f w_i \, dt$$

Herein τ is a factor which represents the volumetric increase due to the temperature increase. If it be assumed that the cylinder contents and the intake air do not mix at first, then, with m as the exponent of the polytropic change in the cylinder,

$$\frac{dp_i}{dt} \frac{1}{p_i} = - \frac{m}{V_z} \left(\frac{dV_2}{dt} - \frac{dV_1}{dt} \right) \quad \text{and} \quad \frac{dp_i}{dt} = \frac{m p_i}{V_z} \left(f w_i \tau - V_h \frac{dz}{dt} \right)$$

Now, however, the intake air and the cylinder contents do mix, and it is to be investigated as to whether the pressure in the cylinder is changed by this mixing. Let T_i denote the temperature of the inflowing air and T_z the temperature of the rest of the cylinder contents. The internal energy must be the same before and after mixing, if the latter is accomplished in so short a time that the motion of the piston and the heat transfer can be disregarded. It is

$$c_v \frac{dv_z p_z T_i}{T_i R} + \frac{c_v (V_z - dv_z) p_z T_z}{T_z R} = c_v \frac{V_z (p_z + dp_z) (T_z + dT_z)}{(T_z + dT_z) R}$$

This yields $V_z dp_z = 0$, showing that the pressure is not affected.

In order to be independent of the absolute dimensions and to obtain laws of similitude, the following ratios are introduced.

$$y = x/l, \quad w = u \frac{V_h 6n}{f}, \quad t = \frac{\alpha}{6n}$$

l is the length of the intake pipe, n the revolution speed and α the crank angle, while u indicates the percentage of the cylinder displacement per crank angle covered by a cross section moving at the velocity w in the intake pipe. On introducing these values into the equations, we obtain

$$-\frac{\partial p}{\partial t} \frac{l}{f} = \rho u \frac{\partial u}{\partial x} \frac{36 n^2 V_h^2}{f^2 l} + \rho \frac{\partial u}{\partial x} \frac{36 n^2 V_h}{f} + \rho u^2 \frac{36 n^2 V_h^2}{f^2 l}$$

$$\frac{\partial \rho}{\partial \alpha} 6n + \rho \frac{\partial u}{\partial \alpha} \frac{6n V_h}{f l} + \frac{\partial \rho}{\partial x} \frac{6n V_h}{f l} = 0$$

$$6n \frac{dp_i}{d\alpha} = \frac{m p_i}{z_0 + z} \left(\frac{f 6n V_h}{V_h f} u \tau - \frac{dz}{d\alpha} 6n \right)$$

$$\frac{u^2}{2g} = \frac{f^2}{36 V_h^2 n^2} \int_{p_0}^{p_a} v dp$$

With $a = \frac{f}{36 n^2 V_h l}$, $b = \frac{V_h}{f l}$, $R = r l$, we finally obtain:

$$- a \frac{\partial p}{\partial x} = b \rho u \frac{\partial u}{\partial x} + \rho \frac{\partial u}{\partial \alpha} + \rho b R u^2$$

$$\frac{\partial p}{\partial \alpha} + \rho \frac{\partial u}{\partial x} b + b u \frac{\partial \rho}{\partial x} = 0$$

$$\frac{dp_i}{d\alpha} = \frac{m p_i}{z_o + z} \left(u_i \tau - \frac{dz}{d\alpha} \right)$$

$$\frac{u^2}{2g} = \frac{a}{b} \int_{p_o}^{p_a} v dp$$

The relations are therefore determined by a , b and R , when the possibility of changing the beginning and end of the intake are disregarded. Like values of these quantities correspond to the same volumetric efficiency, while a and b can be restored to their ordinary values. If the mean velocity in the intake pipe is designated by w_m , we then have, according to the customary equation for the dimensioning of the control sections,

$$w_m = \frac{F c}{f}$$

in which F is the piston area and c the mean piston velocity. We therefore obtain

$$a = \frac{1}{1080 w_m n l}, \quad b = \frac{30 w_m}{l n}$$

The value of R depends on the roughness of the inner surface of the pipe, on the Reynolds Number, and on the dimensions of the pipe.

Integration of the differential equations is impossible. One is therefore dependent on the point to point determination of the velocity and pressure by approximation methods.

Two-Stroke-Cycle Engines with Crankcase Scavenging

Due to the great detrimental space of the scavenging pump (some 300 to 500 per cent), the pressure fluctuations in the intake pipe and in the cylinder are small. The density of the air in the intake pipe may therefore, without great error, be assumed to be constant. We then obtain

$$-a \frac{\partial p}{\partial x} = \rho \frac{\partial u}{\partial \alpha} + \rho b R u^2$$

If we then integrate with the limits $x = 0$ and $x = l$, we obtain

$$a (p_a - p_i - p_v) = \rho \frac{\partial u}{\partial \alpha} + \rho b R u^2.$$

For the orifice, we have

$$a (p_o - p_a) = \frac{b \rho u^2}{2}$$

By addition we obtain

$$a (p_o - p_i - p_v) = \rho \frac{\partial u}{\partial \alpha} + u^2 \rho b (R + 0.5),$$

in which p_v is the portion of the intake-valve resistance independent of the velocity. According to previous formulas, we have

$$\frac{d p_i}{d \alpha} = \frac{m p_i}{z_o + z} \left(u \tau - \frac{dz}{d \alpha} \right).$$

In view of the small pressure fluctuations, we can approximately replace p_i by p_o . The volume of the crankcase can be represented by the mean value during the intake. The air in the intake pipe must also be taken into consideration, since it participates in the shock absorption. The approximately linear pressure decrease in the intake pipe justifies the addition of half the volume of the intake pipe. We therefore have

$$x' = \frac{f l}{2 V_h} + z_o + 0.5$$

u is replaced by $ds/d\alpha$. Hence s is the inflowing quan-

tity of air in percentages of the piston displacement. We obtain

$$p_i = \frac{m p_o}{x'} (s \tau - z) + C$$

$p_i = p_o$ for $s \tau = z$. Hence

$$a (p_o - p_i - p_v) = \frac{a m p_o}{x'} (z - s \tau) - a p_v$$

and, in conjunction with the previous equation,

$$\frac{d^2 s}{d\alpha^2} = \frac{a m p_o}{x' \rho} (z - s \tau) - \frac{a p_v}{\rho} - \left(\frac{ds}{d\alpha} \right)^2 b (R + 0.5)$$

With

$$z_1 = \frac{p_v x'}{m p_o}, \quad A = \frac{a m p_o}{\rho x'} = \frac{m f g p_o}{36 n^2 V_h x' \gamma l}, \quad B = \frac{V_h}{f l} (R + 0.5)$$

we obtain the final form

$$\frac{d^2 s}{d\alpha^2} = A (z - z_1 - s \tau) - B \left(\frac{ds}{d\alpha} \right)^2$$

s is obtained by point to point integration. The resistance of the valve retards the beginning of the intake and causes a pressure reduction of p_v in the crankcase. Hence s must be multiplied by the factor

$$\frac{p_a - p_v}{p_a}$$

Allowance for the retardation of the beginning of the intake due to the inlet port has to be made by increasing the value of z_1 . R also receives the share of the valve resistance dependent on the velocity.

The integration is sufficiently accurate when the change in s for a point 5 to 10° distant is calculated from the first and second differential quotients in a point. For automatic intake valves the integration must be continued till the motion of the air columns is reversed. In Figure 2 the volumetric efficiency is represented as a function of A and B , z_1 being 25 per cent. The maximums of the curves $B = \text{constant}$ all lie approximately at $A = 8/10000$. The curves are flat near the max-

imum and in the region of the large A values. Slight variations in the revolution speed therefore have no appreciable effect on the volumetric efficiency in a correctly dimensioned intake pipe. For values of z_1 in the vicinity of 25 per cent, the volumetric efficiency can be estimated by adding to the diagram values the difference between z_1 and 25 per cent when z_1 is smaller and subtracting when z_1 is greater than 25 per cent.

In Figures 3, 4, and 5 the velocities and the inflowing quantities of air are plotted against the crank angle. It is seen that the air column comes to rest proportionately later, after the dead center, the smaller a is and the larger therefore the relative vibration period of system I. In Figure 6 the difference in time between the dead center and the end of the air motion is plotted against A . It is hardly affected by the damping B . The amplitude of the vibration and consequently also the magnitude of the supercharge is, on the contrary, considerably affected by B .

This calculation method can also be used, when the piston controls the intake port. Figure 7 is a diagram of this arrangement. The upward stroke of the piston produces a negative pressure in the crankcase. The air in it, which acts as a shock absorber, is thereby expanded. When the intake port is opened, the air in the intake pipe is strongly accelerated and the crankcase is considerably supercharged. On the closing of the intake port the air should come to rest immediately. If the intake port is closed too soon, not all of the kinetic energy is utilized; if too late, the air column swings back and a portion of the air escapes. The damping is here generally less than with intake valves. This has a favorable effect on the magnitude of the supercharge.

The integration should be continued up to the closing of the intake port. The damping depends on the opening of the intake port and hence on the position of the piston. The value of B therefore corresponds to the position of the piston. Let f_s represent the controlled cross section. Then for

$$f_s > f, \quad B = \frac{V_h}{f l} (R + 0.5)$$

and for

$$f_s < f, \quad B = \frac{V_h}{f l} \left[R + 0.5 \left(\frac{f}{f_s} \right)^2 \right]$$

With ψD as the width of the intake port, we have $f_s = (\sigma_a - z) \psi D s$, in which σ_a is the length of the port. Hence B can be determined for all positions of the piston, either when R is determined experimentally or is calculated from formulas for the resistance of the pipe.

Figure 8 shows the relation between A , B , and s for a special case. The values of B are valid for the dead-center position of the piston. The engine had a piston displacement of $V_h = 2.03$ liters (123.9 cu.in.) and an intake pipe of 2 inches inside diameter. r was found to be 0.26 per meter. The intake port had a width of 0.68 D and a length of 23 per cent.

The maximum of all the curves $B = \text{constant}$ lies approximately at $A = 10.5/10000$. It is noteworthy that, for $A = 21/10000$, all the curves pass approximately through one point. The curves are flat near the maximum, so that slight deviations of the revolution speed from the most favorable value are here without special influence on the volumetric efficiency. In Figure 9 the velocity and the quantity of the inflowing air are plotted against the crank angle for $A = 5, 10, 30/10000$ and $B = 0.6$. From the course of the velocity curves it is obvious that: at $A = 5/10000$, the vibration is prematurely interrupted; at $A = 10/10000$, the motion of the air in the pipe just at the closing of the intake port ceases independently of it; and, at $A = 30/10000$, the closing occurs so late that some of the air escapes through the intake pipe.

Every port length corresponds to a definite opening period of the port. In the present case it is 104° . The most favorable A for every opening period can be determined approximately by a simple consideration. From the velocity curves it is obvious that the best vibration period of system I is approximately equal to double the period of opening. The dampings but slightly affect the vibration period. For the undamped vibration, the vibration period is

$$T_a = \frac{2\pi}{\sqrt{A}}$$

With α as the opening period, we have

$$A = \left(\frac{\pi}{\alpha}\right)^2$$

In order to show the character of the dependence of the volumetric efficiency on the opening period and on A , the volumetric efficiency was determined for $B = 0.4$ as a function of both values. The experimental engine with the 2-inch intake pipe was used for this purpose. The curves shown in Figure 10 were obtained. The values for the best A obtained by the above methods are marked by vertical dashes. They agree well with the maximums of the curves.

Four-Stroke-Cycle Engines

In these the pressure variations are generally quite large. For more accurate calculations, the density variation of the air in the intake pipe must therefore be taken into account. An approximation method for determining the pressure and velocity can be obtained by assuming the dependence of these values on x . Instead of partial equations, we then obtain ordinary differential equations, which can be integrated from point to point. For short pipe sections, in rough approximation also for the whole pipe, it may be assumed that the pressure and velocity are linearly dependent on x . The greater the number of the pipe sections, the more laborious the calculation, but the more accurate the result. We put

$$p = p_a + \pi x, \quad u = u_a + \gamma x$$

and then obtain, from the previously developed equations,

$$-a \int_{p_0}^{p_0 + \pi} \frac{dp}{\rho} = b \int_0^1 (u_a + \gamma x) \frac{\partial u}{\partial x} dx + \frac{\partial u}{\partial \alpha} \int_0^1 dx + b R \int_0^1 (u_a + \gamma x)^2 dx$$

$$\frac{du_a}{d\alpha} + \frac{1}{2} \frac{d\gamma}{d\alpha} = -a \int_{p_0}^{p_0 + \pi} \frac{dp}{\rho} - b \left[R u_a^2 + (R+1) u_a \gamma + \gamma^2 \left(0.5 + \frac{R}{3} \right) \right]$$

Furthermore, with m_r as exponent for the change in condition in the pipe, by integration from

$$\frac{\partial \rho}{\partial \alpha} + \rho \frac{\partial u}{\partial \alpha} b + b u \frac{\partial \rho}{\partial x} = 0$$

we obtain

$$b \int_0^1 \frac{\partial u}{\partial x} dx = - \frac{1}{m_r} \int \frac{1}{p} \frac{\partial p}{\partial x} dx - \frac{b}{m_r} \int \frac{u}{p} \frac{\partial p}{\partial x} dx$$

If the above expressions are introduced for u and p , we then have:

$$m_r \gamma b = \ln\left(1 + \frac{\pi}{p_a}\right) \left[-\frac{dp_a}{d\alpha} \frac{1}{\pi} + \frac{d\pi}{d\alpha} \frac{p_a}{\pi^2} - b u_a + \frac{p_a}{\pi} \gamma b \right] - \frac{d\pi}{d\alpha} \frac{1}{\pi} - b \gamma$$

With $\frac{dp_i}{d\alpha} = \frac{dp_a}{d\alpha} + \frac{d\pi}{d\alpha}$ we obtain: $\frac{d\pi}{d\alpha} p_o b \gamma f_1\left(\frac{\pi}{p_a}\right) + \left(\frac{dp_i}{d\alpha} + u_a b \pi\right) f_2\left(\frac{\pi}{p_a}\right)$

in which

$$f_1\left(\frac{\pi}{p_a}\right) = \frac{\frac{m_r + 1}{\ln\left(1 + \frac{\pi}{p_a}\right)} \frac{\pi}{p_a} - 1}{1 + \frac{p_a}{\pi} - \frac{1}{\ln\left(1 + \frac{\pi}{p_a}\right)}}, \quad f_2\left(\frac{\pi}{p_a}\right) = \frac{1}{1 + \frac{p_a}{\pi} - \frac{1}{\ln\left(1 + \frac{\pi}{p_a}\right)}}$$

$f_1\left(\frac{\pi}{p_a}\right)$ and $f_2\left(\frac{\pi}{p_a}\right)$ are represented in Figure 11.

These equations are valid for each of the n parts of the pipe. There are, therefore, n equations for the pressure and just as many for the velocity. Every series of $n - 1$ equations is obtained by equating the differential quotients of the abutting pipe-section ends. With I, II n as the pipe sections, we obtain:

$$\left(\frac{du_a}{d\alpha}\right)_I = \left(\frac{du_a}{d\alpha}\right)_{II} + \left(\frac{d\gamma}{d\alpha}\right)_{II}$$

$$\left(\frac{du_a}{d\alpha}\right)_{n-I} = \left(\frac{du_a}{d\alpha}\right)_n + \left(\frac{d\gamma}{d\alpha}\right)_n$$

$$\left(\frac{dp_i}{d\alpha}\right)_I = \left(\frac{dp_o}{d\alpha}\right)_{II} + \left(\frac{d\pi}{d\alpha}\right)_{II}$$

$$\left(\frac{dp_i}{d\alpha}\right)_{n-I} = \left(\frac{dp_o}{d\alpha}\right)_n + \left(\frac{d\pi}{d\alpha}\right)_n$$

The equation for the velocity at the entrance applies unchanged, as likewise the one for the change in condition in the cylinder. We thus obtain $2n$ equations, from which the differential quotients can be calculated for the given conditions. If the Euler approximation method is used, we obtain, for a small time interval,

$$\Delta u_a + 0.5 \Delta \gamma = -a \Delta \alpha \int \frac{dp}{\rho} - b \Delta \alpha [R u_a^2 + (R+1) u_a \gamma + (0.5 + 0.33 R) \gamma^2]$$

$$\Delta \pi = p_a \quad b \quad \gamma \quad f_1 \left(\frac{\pi}{p_a} \right) \Delta \alpha + (\Delta p_i + u_a b \pi \Delta \alpha) f_2 \left(\frac{\pi}{p_a} \right)$$

$$\Delta p_a = \Delta p_i - \Delta \pi$$

The accuracy can be increased by introducing into the equations the mean values of those quantities which vary greatly during the interval.

$$\text{Instead of } u_a \text{ we put } u_a + \frac{\Delta u_a}{2}$$

$$\text{" " } \gamma \text{ " " } \gamma + \frac{\Delta \gamma}{2}$$

$$\text{" " } \pi \text{ " " } \pi + \frac{\Delta \pi}{2}$$

$f_1 \left(\frac{\pi}{p_a} \right)$ and $f_2 \left(\frac{\pi}{p_a} \right)$ can be assumed to remain unchanged.

We finally obtain

$$\Delta p_i = \frac{m p_i}{z_0 + z} \left[(u_a + \gamma) \tau - \frac{dz}{d\alpha} \right] \Delta \alpha + \frac{m p_i \tau}{z_0 + z} (\Delta u_a + \Delta \gamma) \frac{\Delta \alpha}{2}$$

$$\Delta \pi \left[1 - \frac{b u_a}{2} f_2 \left(\frac{\pi}{p_a} \right) \Delta \alpha \right] = b p_a \gamma f_1 \left(\frac{\pi}{p_a} \right) \Delta \alpha + (p_i + b u_a \pi \Delta \alpha) f_2 \left(\frac{\pi}{p_a} \right) +$$

$$+ \Delta \gamma b p_a f_1 \left(\frac{\pi}{p_a} \right) \frac{\Delta \alpha}{2} + \Delta u_a b \pi f_2 \left(\frac{\pi}{p_a} \right) \frac{\Delta \alpha}{2}$$

$$\Delta u_a \left[1 + \Delta \alpha b u_a f \left(\frac{\gamma}{u_a} \right) \right] + 0.5 \gamma = -a p_0 v_0 g \beta \left(\frac{\pi}{p_0} + \frac{\Delta \pi}{2 p_0} \right) \Delta \alpha$$

$$- b u_a^2 f \left(\frac{\gamma}{u_a} \right) \Delta \alpha$$

$$\text{in which } f \left(\frac{\gamma}{u_a} \right) = R + (R+1) \frac{\gamma}{u_a} + (0.5 + 0.33 R) \left(\frac{\gamma}{u_a} \right)^2$$

At the mouth of the pipe,

$$u_a \Delta u_a + \frac{\Delta u_a^2}{2} = - \frac{a}{b} p_o v_o g \beta' \frac{\Delta p_a}{p_o}$$

In the equations

$$\int_{p_a}^{p_i} \frac{dp}{\rho} = p_o v_o g \frac{p_i - p_a}{p_o} \beta, \quad \int_{p_{an}}^{p_o} \frac{dp}{\rho} = p_o v_o g \frac{(p_o - p_{an})}{p_o} \beta'.$$

From these equations Δu_o , $\Delta \gamma$, Δp_a and $\Delta \pi$ can be calculated for every portion of the pipe.

From the equations, it must follow that the propagation of any change of condition in the pipe proceeds with the velocity of sound. If the air in the pipe is at rest and the pressure is changed at its cylinder end, we have, for small pressure fluctuations,

$$f_1 \left(\frac{\pi}{p_a} \right) = 2 m_r, \quad f_2 \left(\frac{\pi}{p_a} \right) = 2, \quad \beta = 1.$$

After $\Delta \alpha$ crank degrees the disturbance lm has reached the point I. This value l is introduced into the expressions for a and b . Then Δu_a and Δp_a must be 0, since, though the disturbance has reached the point I, the condition there has not yet been changed. We obtain

$$\Delta \pi = 2 \Delta p_i + 2 m_r \Delta \gamma b p_a \frac{\Delta \alpha}{2}$$

Now, $\Delta p_i = \Delta \pi + \Delta p_a$ i.e., $\Delta p_i = \Delta \pi$. Therefore,

$$- \Delta \pi = m_r \Delta \gamma b p_a \Delta \alpha$$

and consequently,

$$a b p_a v_a g m_r \Delta \alpha^2 = l$$

If the expressions for a , b and $\Delta \alpha$ are introduced, we obtain

$$c = \frac{l}{\Delta t} = \sqrt{p_a v_a g m_r}.$$

which is the velocity of sound.

The following example illustrates the application of the method of calculation. The course of the pressure and

velocity are to be calculated for a four-stroke engine with $D = 209$ mm (8.23 in.), $s = 311$ mm (12.24 in.), $n = 800$, and a brass intake pipe having a diameter of $d = 63.5$ mm (2.5 in.) and a length of $l = 1.05$ m (41.34 in.). Let the number of parts for the calculation be 3 and the time interval be $\Delta\alpha = 5^\circ$. The resistance of the pipe is estimated from Schüle's data, and it is assumed that $m = 1.2$ and $\tau = 1.1$.

The following equations are obtained:

For the change of condition in the cylinder,

$$\Delta p_i = \frac{1.2 p_i}{0.07 + z} [5.5 (u_a + \gamma) - \Delta z] + \frac{3.3 p_i}{0.07 + z} (\Delta u_a + \Delta \gamma)$$

For the flow in the pipe sections,

$$\begin{aligned} \Delta \pi \left[1 - 25 u_a f_1 \left(\frac{\pi}{p_a} \right) \right] &= 50 p_a \gamma f_1 \left(\frac{\pi}{p_a} \right) + \Delta p_i f_2 \left(\frac{\pi}{p_a} \right) + 50 u_o \pi f_2 \left(\frac{\pi}{p_a} \right) \\ &+ \Delta \gamma 25 p_a f_1 \left(\frac{\pi}{p_a} \right) + \Delta u 25 \pi f_2 \left(\frac{\pi}{p_a} \right) \\ \Delta u \left[1 + 25 u_a f \left(\frac{\gamma}{u_a} \right) \right] + \frac{\Delta \gamma}{2} &= -15 \beta \pi 10^{-7} - 7.5 \beta \Delta \pi 10^{-7} - 50 u_a^2 f \left(\frac{\gamma}{u_a} \right) \end{aligned}$$

For the mouth of the pipe,

$$u_a \Delta u_a + \frac{\Delta u_a^2}{2} = -3 \times 10^{-8} \beta \Delta p_a$$

The values of Δu_a , $\Delta \gamma$, $\Delta \pi$, and Δp_i can be calculated from the above equations without great difficulty.

Figures 12 and 13 show the course of the velocity and pressure. The intake begins at 15° . The change of condition reaches the entrance at 30° . The previously almost linear course of the pressure and velocity is thereby changed. Vibrations are started in the pipe and are propagated from its mouth toward the cylinder.

The calculation for the whole intake stroke would not be difficult, but very tedious. It is simpler, though less accurate, to assume only one section of pipe. We then obtain the pressure and velocity courses shown in Figures 14 and 15. Thereby it is assumed, just as before, that the contents of the pipe are at rest at the beginning of

the intake stroke, which is the case when the vibrations of system II are small. This conclusion is substantiated by the experimental data.

The pressure in the cylinder falls until 45° , then rises again and reaches 1 atmosphere at 145° and its maximum value of 1.2 atm. at 200° , that is, a little after the dead center. In Figure 16, the pressure is plotted against the crank angle. In one case the calculations were based on $m = 1.2$ and once on $m = 1.4$. It is seen that the exponent of the change in condition only slightly affects the result. The measured curve was taken from a published diagram.* Perfect agreement was not to be expected, since the resistance in the intake system of the experimental engine was not accurately known. Moreover, the the pipes differed a little in length.

Figure 17 gives the velocities at the entrance to the cylinder. If we plot $\frac{u_a + \gamma}{v}$, we obtain by planimetry the weight of the air taken in, namely, $1.225 V_h$ kilograms per intake stroke. Under external conditions of 1 atm. and 15°C (59 F.), $\eta_l = 103$ per cent. The temperature of the charge at the end of the intake is therefore 58°C (136.4°F.).

We will now compare the energy absorbed in supercharging by an intake pipe with that absorbed in supercharging by a blower and with the theoretically necessary energy. For calculating the latter, imagine the piston stroke so greatly increased that, in compression up to the actual cylinder volume, the supercharging pressure p_1 is attained. With p_0 as the intake pressure, the work to be performed is

$$A = \frac{k}{k-1} V_h p_1 \left[1 - \left(\frac{p_0}{p_1} \right)^{\frac{k-1}{k}} \right] - (p_1 - p_0) V_h$$

Applied to the previously calculated example, we obtain $A = 120 V_h$ mkg. The adiabatic work of the blower is

$$A_{ga} = \frac{k}{k-1} V_h p_1 \left[1 - \left(\frac{p_0}{p_1} \right)^{\frac{k-1}{k}} \right]$$

With 80 per cent efficiency this yields $A_g = 2530 V_h$ mkg.

*Joachim, "Forschungen über Schwerölmotoren in den Ver. Staaten. Dieselmotoren V, V.D.I., 1932, pp. 75-82.

A part of this is regained through the piston. The velocity at the valve and the resulting necessary pressure drop can be approximately determined from the continuity equation

$$\frac{u^2}{2g} = \frac{a}{b} \int v \, dp.$$

For 80 mm (3.15 in.) valve diameter, we obtain a negative pressure of

$$\Delta p = 6.4 u^2$$

and from this the intake curve represented in Figure 18 with a mean pressure of 11.65 kg/m² (2.39 lb./sq.ft.). The energy recovery is 1,650 V_h mkg, a difference of 880 V_h mkg. The mean negative pressure in the intake pipe is 1,600 kg/m² (327.7 lb./sq.ft.); hence the energy consumption is 1,600 V_h mkg. If the energy consumption, due to the acceleration at the valve, is subtracted in both cases, we obtain the increase in energy consumption due to supercharging:

Blower, 530 V_h mkg

Intake pipe, 250 V_h mkg

The efficiency, i.e., the ratio of the theoretically necessary energy to that actually required is 23 per cent for the blower and 9.5 per cent for the intake pipe. It is low in both cases, but considerably lower for the intake pipe than for the blower. It has but little effect on the engine efficiency.

TEST INSTALLATION

Figure 19 shows the test installation. The engine has a bore of 120 mm (4.72 in.) and a stroke of 180 mm (7.09 in.). It is driven by an electric motor with revolution speed variable between 400 and 700. For facilitating the escape of the air in scavenging, the pipe to the air-measuring device is attached directly to the cylinder, the outlet being closed with a screw cap. The air can enter the scavenge pump through automatic valves and through an opening controlled by the piston. The clearance space in the scavenging pump can be varied by varying the amount of water in the tank b.

The air-measuring device consists of an equalization tank *a* and the gasometer *g*. Between the two is a valve *v*, which is closed during the test. On the gasometer there is a drum *t* covered with paper along which moves an index *z*. At every revolution a spark jumps from the index to the drum and punctures the paper. From the mean distance between the marks thus obtained, the volume of air drawn in at each stroke can be determined. The device was made four years ago, was used in many tests and worked simply and accurately.

For determining the resistance of the intake pipe, the pressure difference between the mouth of the intake pipe and the cylinder was measured with a uniform air flow. In the tests the air speed was changed, as likewise the piston position for controlled intake orifice. The air was furnished by a supercharger and its quantity was determined by a pressure disk.

TEST RESULTS

Intake Port Controlled by the Piston

The clearance space of the scavenging pump was 500 per cent. The volumetric efficiencies were determined for intake pipes with diameters of 1-3/4, 2, and 2-5/8 inches and lengths of 0 to 3.5 m (137.8 in.) at revolution speeds of 400, 500, 600 and 700. The longitudinal gradations were, in general, 0.5 m (19.7 in.), but 0.1 m (3.94 in.) for the telescopically extensible 2-inch pipe. The length of the intake port was changed in four gradations by shortening the piston. The heavy lines in Figures 20-22 represent the efficiency as a function of the pipe length. It is obvious that there is considerable increase in the volumetric efficiency and that relatively short pipes are required for this purpose. In Figure 21, as a result of the small length gradations, we can clearly recognize the effect of the vibrations of system II. This increases with the pipe length. It is still small in the practically important region about the maximum. In addition to these curves for 104° opening period, others were determined for 95°, 88°, and 78°. For 95° the maximum is at approximately the same height, while for 88° and 78° it was below that of the curves for 104°. Since these curves show nothing essentially new, the tests were not repeated. It is noteworthy that, through a favorable

combination of the vibrational systems I and II, the curves for the 2-5/8-inch pipe, 95° opening period, $n = 500$, climb in a point to 137 per cent.

The experimental results will now be compared with the mathematical results. For the change of condition in the cylinder, an exponent of 1.3 was obtained from weak-spring diagrams made with Maishak's bar-spring indicator. The B values were obtained by resistance tests.

The length l of the intake pipe must be increased for the calculation by the amount Δl , since not only the air in the intake pipe, but also the dynamic effect of the air in the passage at the entrance to the intake pipe and of the downstream flow in the cylinder must be taken into account. The increase amounted to 0.5 m (19.7 in.) for the 2-5/8-inch pipe, 0.3 m (11.8 in.) for the 2-inch pipe, and 0.2 m (7.87 in.) for the 1-3/4-inch pipe. It is approximately proportional to the cross section of the pipe. If we imagine the whole system replaced by pipes of different cross sections, we obtain the scheme represented in Figure 23 with f, l as the intake pipe. The pressure drop between the crankcase and the outside air, due to the acceleration, is

$$\Delta p_b = \frac{\gamma}{g} (lb + l_1 b_1 + l_2 b_2).$$

According to the law of continuity we have

$$b_1 = b \frac{f}{b_1'}, \quad b_2 = b \frac{f}{b_2'}$$

and consequently,

$$\Delta p_b = \frac{\gamma}{g} b \left(l + l_1 \frac{f}{f_1} + l_2 \frac{f}{f_2} \right) \quad \text{and} \quad \Delta l = f \left(\frac{l_1}{f} + \frac{l_2}{f} \right)$$

This explains the proportionality between the increase in length and the cross section of the pipe.

The vibrations of system II are pipe vibrations. During the intake they correspond to those of an open pipe; between the intake strokes, to those of a closed pipe. If the vibration period is represented by its ratio to the time required for revolution, we obtain

$$e = C \frac{l}{n l}$$

in which C is a constant. The effect of the vibrational system II depends on l and consequently on nl . If the volumetric efficiency is plotted against nl , the maximums and minimums of the undulating curves must coincide, independently of other relations. In Figure 24 this was done for the curves in Figure 21 and thereby the ideal pipe length $(l\Delta + l)$ was introduced into ln . The above-mentioned coincidence is recognized at I, II, and III.

We can now calculate A . For the 2-inch pipe the volumetric efficiency can then be taken directly from Figure 8, since $z_1 = 20$ per cent corresponds to the conditions under consideration (20 per cent until the closing of the intake ports by the piston rings). We obtain the lightly drawn curves in Figure 22, whose good agreement with the measurements in the practically important region of the maximum demonstrates the availability of this method of calculation. With the longer pipes there are deviations due to the vibrations of system II, which are disregarded in the calculation. This does not explain the disagreements in the curves for $n = 400$. The interruption of the experiments prevented the further investigation of their causes. This revolution speed is so low that it has no importance for the engine.

In Figure 8, B indicates the piston dead center. The volumetric efficiency is also affected by the values of B in the other positions of the piston. Figure 8 is therefore valid only for the 2-inch pipe and for other pipes in which the values of B for all the piston positions bear the same ratio to the values for the dead center as in the 2-inch pipe. For pipes with larger cross sections in relation to the area of the port, the values of B will increase more and the volumetric efficiency will be less than that derived from Figure 8. Pipes with relatively smaller cross sections yield a higher volumetric efficiency. The effect of the B values for partial opening is not very great, however, as was found by comparison with more accurate calculations. Hence Figure 8 can serve for the approximate determination of the volumetric efficiency in nearly all cases.

With the 2-5/8-inch pipe, it was found from the calculation of individual points that the correct values were obtained from Figure 8 by using 15 per cent higher values of B . With the 1-3/4-inch pipe, the error in the determination of the volumetric efficiency from Figure 8 was negligible. The curves are plotted lightly in Figures

20 and 22 and agree very well with the experimental results.

Figure 25 shows the variation in the volumetric efficiency for small changes in the revolution speed. The longitudinal gradation was here 0.5 m (19.7 in.). The resistance of the pipe differed somewhat from that of the telescopic pipe used for the tests represented in Figure 22, so that the results do not fully agree.

Intake Port with Automatic Valves

In the experiments the clearance space in the scavenging pump was 500 and 800 per cent; the revolution speeds, 400, 500, 600, and 700. The intake pipes had diameters of $1\frac{1}{2}$, 2, and $2\frac{5}{8}$ inches and lengths up to 3.5 m (137.8 in.). The intake valves had steel plates held by springs and opened at 80 mm (3.15 in.) excess water pressure.

In Figures 26-29 the measured volumetric efficiencies of the $1\frac{1}{2}$ and 2 inch pipes are represented for $x = 500$ and 800 per cent with heavy lines. They are considerably lower than for controlled intake orifices and the best pipe lengths are greater. The calculation of the curves was again based on an exponent of the polytrope 1.3, as was also obtained here from the weak-spring diagram.

The dynamic effect of the air column in and immediately before the valve and in the crankcase is also accounted for here by an increase in the pipe length. This increase is 0.3 m (11.8 in.) for the $2\frac{5}{8}$ -inch pipe, 0.2 m (7.87 in.) for the 2-inch pipe, and 0.1 m (3.94 in.) for the $1\frac{1}{2}$ -inch pipe. We can therefore calculate A and take s directly from Figure 2 for all pipes. Here it must be taken into consideration that z_1 , as previously indicated, is different for $x = 500$ and 800 per cent. We obtain the continuous curves, lightly drawn in Figures 26-29, which have the same character as the experimental curves, but are higher than the latter for higher revolution speeds and for large values of x . This is due to the fact that in these cases the crankcase is not completely emptied through the scavenging ports. The cylinder used in these experiments had considerably smaller scavenging-port sections than the one with controlled intake port, so that this error occurred only here. It can depend only on x , n , and the intake air, since the pressure at the

end of the scavenging, and consequently the amount of the residual air is determined only by these quantities. If the error is plotted against the calculated values, the points must lie, independently of the dimensions of the intake pipe, on a curve so shaped that the error increases with increasing quantities of air.

From Figure 30, in which the error was plotted for $x = 800$ per cent, and $n = 500, 600,$ and 700 , it is obvious that the points actually lie with good approximation on a curve of the required form. This and weak-spring diagrams which show that the pressure at the end of the scavenging is higher than the outside pressure in the corresponding cases, demonstrates the correctness of the explanation of the error. The error curve was utilized for the correction of the test results, thus obtaining the dash curves in Figures 26-29, which agree well with the calculated curves.

Further discrepancies between the calculated and measured results are due to the vibrations of system II. The discrepancies are the greater, the longer the pipe and the greater the amount of air in the pipe in relation to the piston displacement. With the best pipe length, the discrepancy is small for the $1\frac{1}{2}$ and 2 inch pipes, but considerable for the 2-5/8-inch pipe.

As suggested in a work of Klüsener,* the undamped pipe vibrations were calculated by means of Fourier series for the 2-inch pipe for $x = 500$ per cent and $n = 400, 500, 600$ and 700 . The calculations were based on the velocity conditions at the intake valve, which were calculated from the vibration of system I. (Figs. 3-5.) Equations for the excess pressure in the pipe for various intake periods were calculated according to the method of Voissel and Klüsener, and the excess pressure was plotted against \ln .

The effect of the pipe length on the volumetric efficiency was determined by means of a telescopic 2-inch pipe in gradations of 0.1 m (3.94 in.). Figure 32 shows the efficiency plotted against \ln . The deviations from the calculated volumetric efficiency curves are caused by the vibrational system II. They were compared with the calculated pressures at the end of the intake. Figure 31 shows them for one revolution speed. The drop in the volumetric efficiency, to be observed even in Klüsener's work, occurs in the field of resonance. The magnitude of

*"Saugrohr und Liefergrad." Dieselmachines V, p. 107.

the deviation caused by vibration II cannot be determined from the equations for the excess pressure, because the damping is disregarded.

SUMMARY

The experimental results show that the vibrations of system I are essential for the volumetric efficiency under practically utilizable conditions. The volumetric efficiency can be determined by the indicated method with an accuracy of a few per cent.

PRACTICAL APPLICATION

Any considerable increase in the volumetric efficiency by intake pipes is possible only with an intake port controlled by the piston. With intake valves the increase in the volumetric efficiency is relatively small, while the necessary length of the intake pipe is greater than in the previously mentioned case and therefore structurally more inconvenient.

Intake Port Controlled by Piston

The dimensions of the intake port are properly similar to that in the experimental engine. In this an opening period from 100 to 110° gave the best results, the corresponding value of A being about $10.5/10000$. Hence the ratio l/f is fixed for given engine dimensions. One chooses only a few values of f corresponding to the ratio, determines s for these values and finds which pipe yields the highest value. In practical applications it is often better to forego a few per cent of s , in order to use a smaller pipe.

Example.— An intake pipe is to be calculated for an engine with crankcase scavenging, in which $D = 150$ mm (5.9 in.), $s = 200$ mm (7.87 in.), $n = 550$. The detrimental space of the scavenging pump is assumed to be 400 per cent, the intake-port dimensions to be $\psi_a = 0.68$, $\sigma_a = 23$ per cent, i.e., the same as in the experiments described. If x' is first estimated at 500 per cent, a value of $f/l = 19/10000$ is obtained with $m = 1.3$ for $A = 10.5/10000$. For constructional reasons, only pipes

not over 2 m (78.7 in.) long are considered. We obtain, for example,

$$l = 1, 1.5, 2 \text{ m}$$

$$d = 50, 60, 70 \text{ mm}$$

The pipe resistance is estimated according to the method of Schüle (Technische Thermodynamik, Vol. I). For calculating Reynolds Number we use the mean value of the velocity, which can be calculated from the inflowing amount of air ($\sim V_h$), from the pipe section, and from the period of opening. We obtain

$$\frac{R}{l} = 10 \text{ inches } \frac{\beta g}{d}$$

Pipe diameter (mm):	50	60	70
Reynolds Number :	170,000	148,000	126,000
β :	11×10^{-8}	11.5×10^{-8}	11.8×10^{-8}
R :	0.22	0.29	0.33

The values of B for the different piston positions, as determined from these data according to what precedes, are:

Crank angle	50	60	70 mm
130	45.0	45.5	46.6
135	3.53	3.43	3.40
140	1.68	1.53	1.28
145	1.32	0.88	0.74
150	1.32	0.67	0.45
155	1.32	0.67	0.43
160	1.32	0.67	0.38
165	1.32	0.67	0.38
170	1.32	0.67	0.38
175	1.32	0.67	0.38
180	1.32	0.67	0.38

(mm \times .03937 = inches)

With respect to the intake port, $z_1 = 20$ per cent. It is possible to calculate s only from point to point. The increase Δs between two points I and II is approximately

$$\Delta s = \left(\frac{d s_I}{d \alpha} \right) \Delta \alpha + \frac{1}{2} \frac{d^2 s_I}{d \alpha^2} \Delta \alpha^2$$

The intervals were 5° . We obtain:

d =	50	60	70 mm	
s =	<u>107.5</u>	<u>124.5</u>	<u>132.1%</u>	volumetric
				efficiency

The largest pipe gives the best result.

The specified length is Δl smaller than l . Δl must be determined experimentally. For the case in question, it can be estimated from the results obtained with the experimental engine. For the same ratio of pipe section to cylinder section, Δl will be approximately proportional to the stroke, which chiefly determines the length of the air column descending in the cylinder.

For the same cross-sectional ratio, a pipe of 63 mm (2.48 in.) corresponds to the 2-inch pipe of the experimental engine.

$$0.3 \times \frac{200}{180} = 0.33 \text{ m}$$

Hence, for 50 mm pipe diameter, $\Delta l = 0.22 \text{ m}$ (8.66 in.)

60 " " " $\Delta l = 0.31 \text{ "}$ (12.2 ")

70 " " " $\Delta l = 0.42 \text{ "}$ (16.54 ")

If we now correct x' and thereby the theoretical pipe length, we obtain the following actual pipe lengths:

d = 50 mm	l = <u>0.84 m</u>
d = 60 "	l = <u>1.19 "</u>
d = 70 "	l = <u>1.41 "</u>

If the pipe is shortened, A and B increase and consequently the volumetric efficiency decreases. If the pipe is lengthened, A and B decrease. The diminution of A decreases the volumetric efficiency, while the diminution of B increases it. It first overcomes the effect of B and then the effect of A. The maximum volumetric efficiency will therefore be reached with a pipe length somewhat longer than the calculated length. The difference is slight, however, under such conditions as existed in the experimental engine.

The volumetric efficiency can be quickly estimated

from Figure 8. With the values for B corresponding to the piston dead center, we have

$$d = 50, \quad 60, \quad 70 \text{ mm}$$

$$s = 105, \quad 123, \quad 136\%$$

These values approximate those previously calculated.

With Intake Valves

The determination of s is here unreliable, since the resistance of the intake valve depends essentially on its construction and can be only roughly estimated. Resistance tests with the valve used in the experimental engine showed that the resistance can be resolved into a constant component and another component which varies with the velocity. The second resistance for the valve tested was equivalent to that of an orifice of 12.6 cm^2 (1.95 sq.in.). For the same air velocity the area f_s of the equivalent opening for the engine of the example was found by calculation to be 20 cm^2 (3.1 sq.in.). The most favorable value of $A = 8/10000$ was obtained from Figure 2. Hence $l/d = 14.5/10000$. This gives

$$d = 50, \quad 60, \quad 70 \text{ mm}$$

$$l = 1.35, \quad 1.88, \quad 2.30 \text{ m}$$

B is calculated from

$$B = \frac{V_h}{f l} \left[R + 0.5 \left(\frac{f}{f_s} \right) \right]$$

Introducing into this R/l , as before, we obtain

$$d = 50, \quad 60, \quad 70 \text{ mm}$$

$$B = 1.07, \quad 0.84, \quad 0.88$$

$$s = \underline{107}, \quad \underline{111}, \quad \underline{110\%}$$

The 60 mm (2.36 in.) pipe is therefore the best.

Δl cannot here be derived from the values of the experimental engine. If they are estimated at 0.15, 0.25, and 0.35 (increasing with the diameter), we obtain the pipe lengths,

$$d = 50, \quad 60, \quad 70 \text{ mm}$$

$$\sim = 1.0, \quad 1.6, \quad 2.0 \text{ m}$$

The volumetric efficiencies are smaller than for controlled openings, and the pipes are longer.

Translation by Dwight M. Miner,
National Advisory Committee
for Aeronautics.

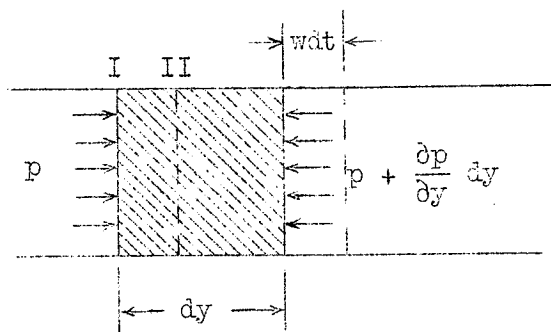


Fig. 1

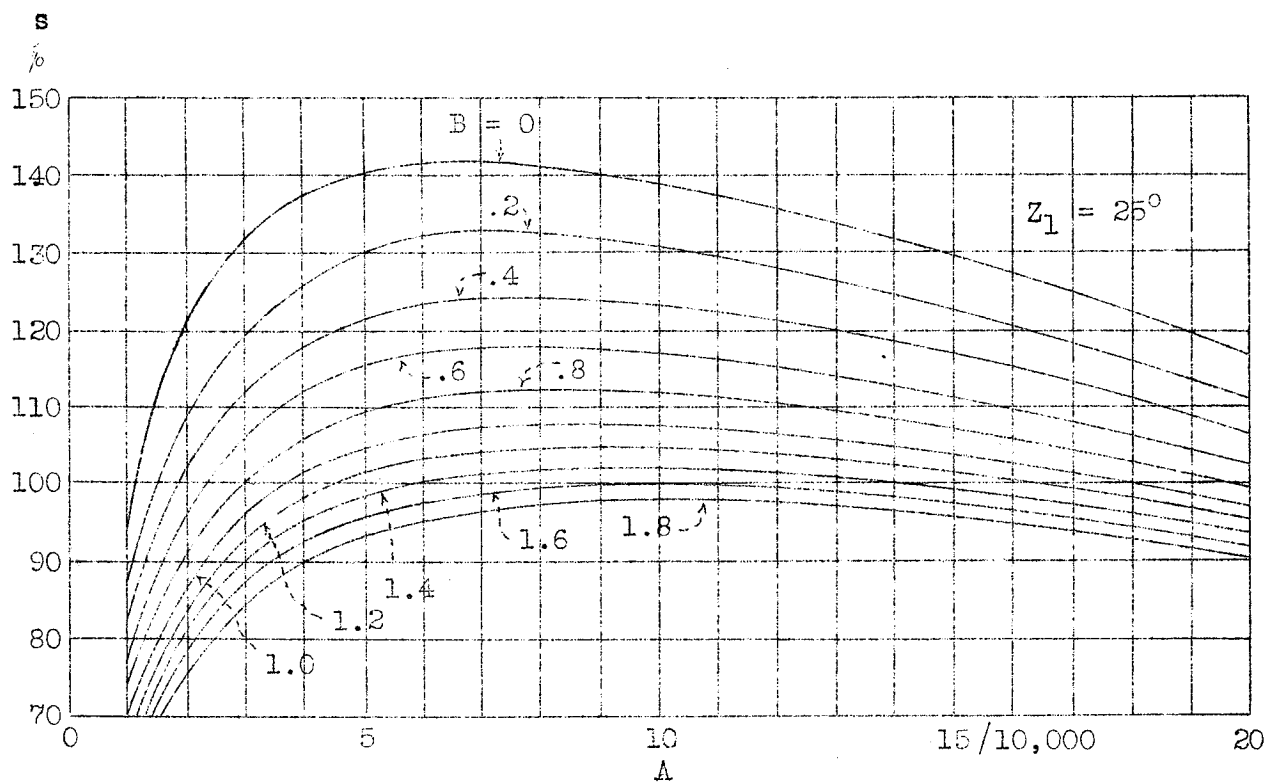


Fig. 2 Volumetric efficiency as a function of A and B. Intake valve.

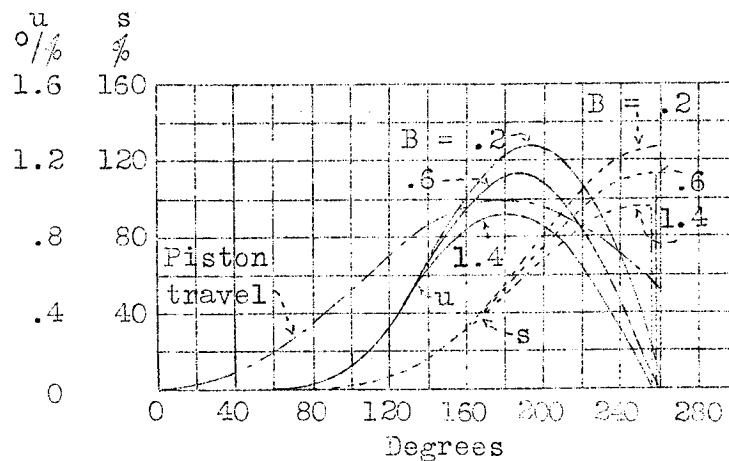


Fig. 3 Air velocities. Quantities of air taken in for $A = 4/10,000$. Intake valve.

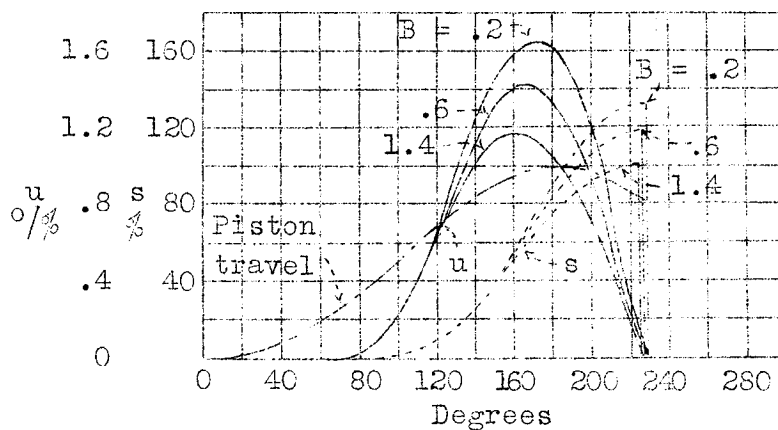


Fig. 4 Air velocities. Quantities of air taken in for $A = 8/10,000$. Intake valve.

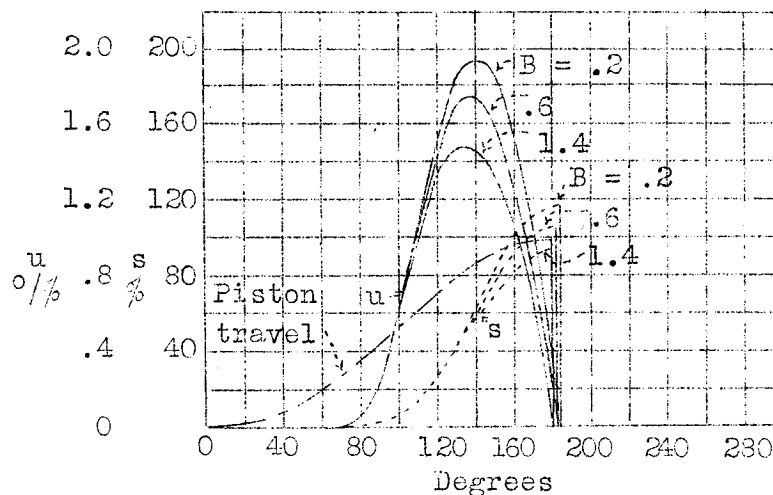


Fig. 5 Air velocities. Quantities of air taken in for $A = 20/10,000$. Intake valve.

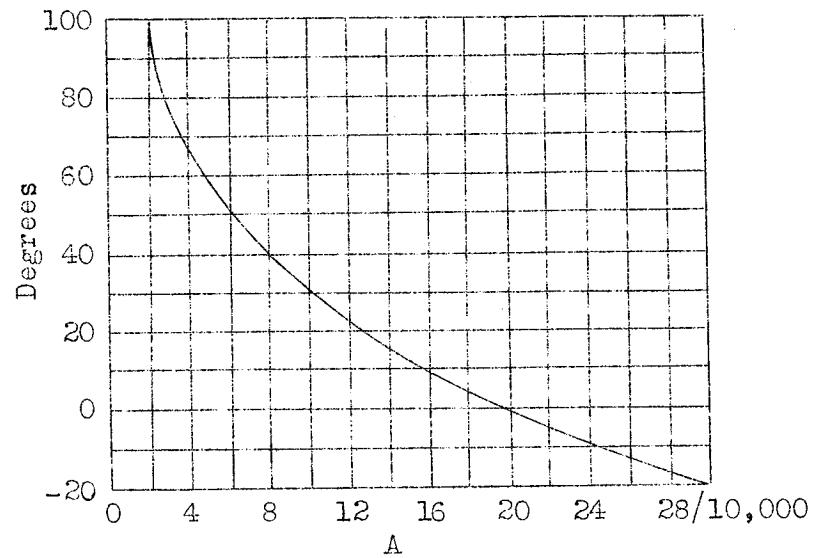


Fig. 6 Time difference between piston dead center and end of air motion in intake pipe (crank angle.)

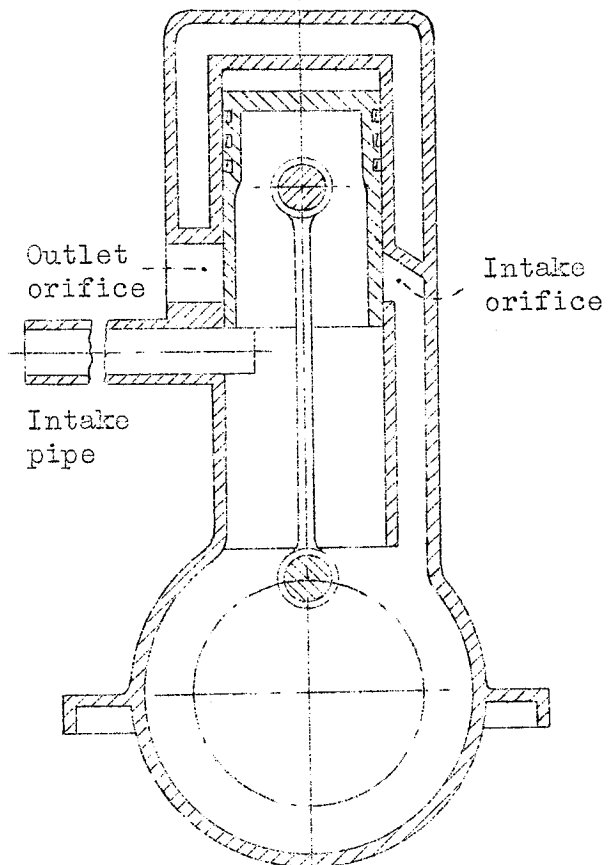
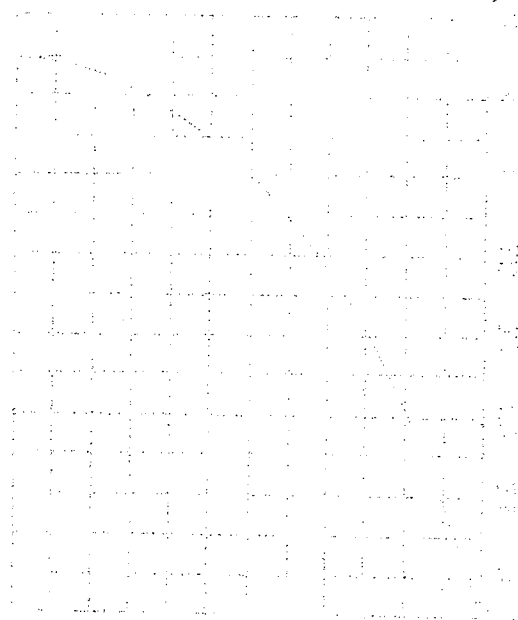


Fig. 7 Diagram of engine with intake port of scavenging pump controlled by piston.



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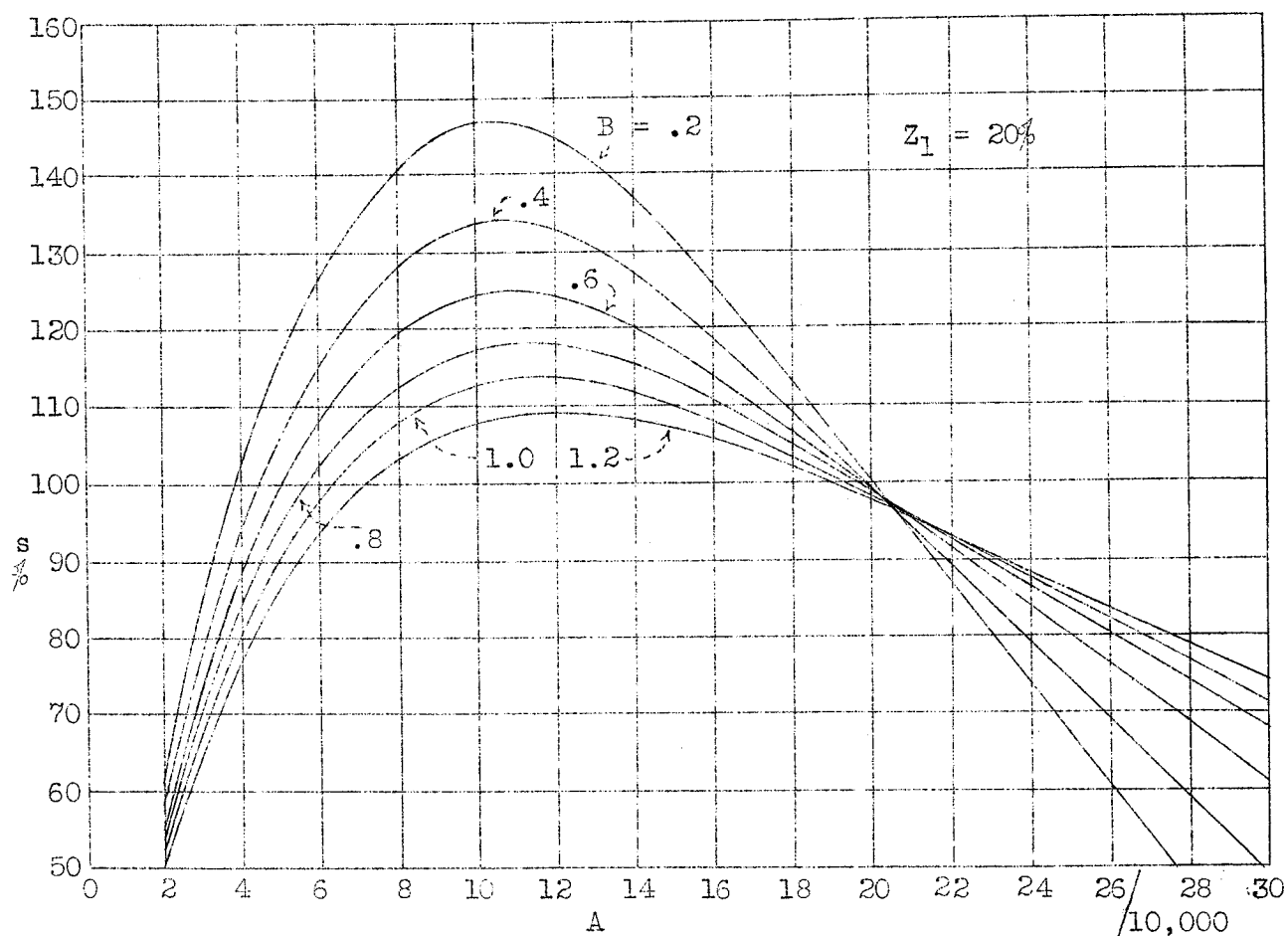


Fig. 8 Volumetric efficiency as a function of A and B with intake orifice controlled by piston. 2 inch intake pipe.

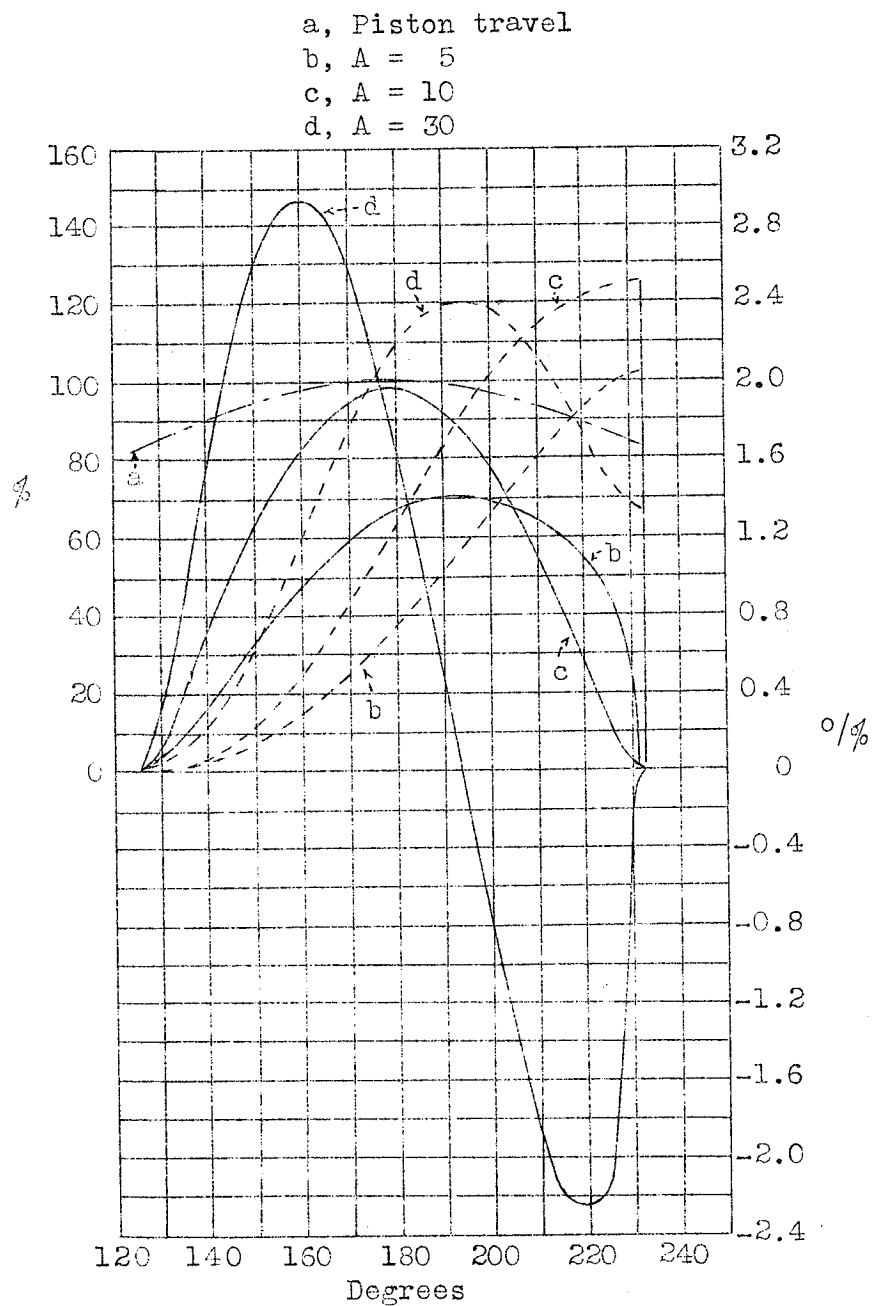
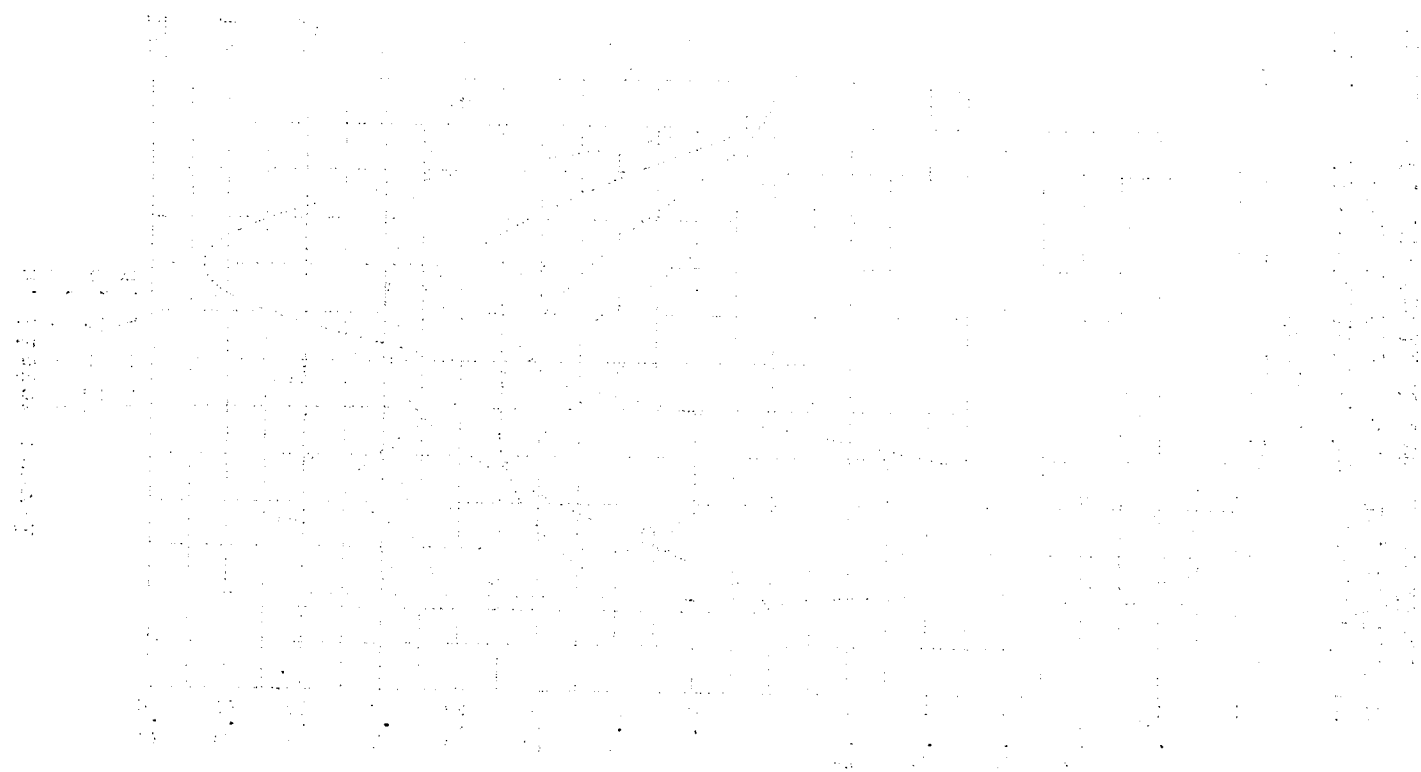


Fig. 9 Air velocities. Quantities of air taken in for $A = 5, 10, 30/10,000$ and $B = 0.6$. Controlled intake orifice.



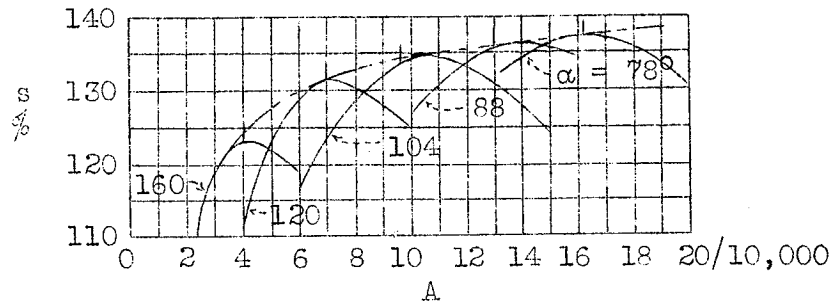


Fig. 10 Effect of duration of opening of intake slot on volumetric efficiency. $B = 0.4$

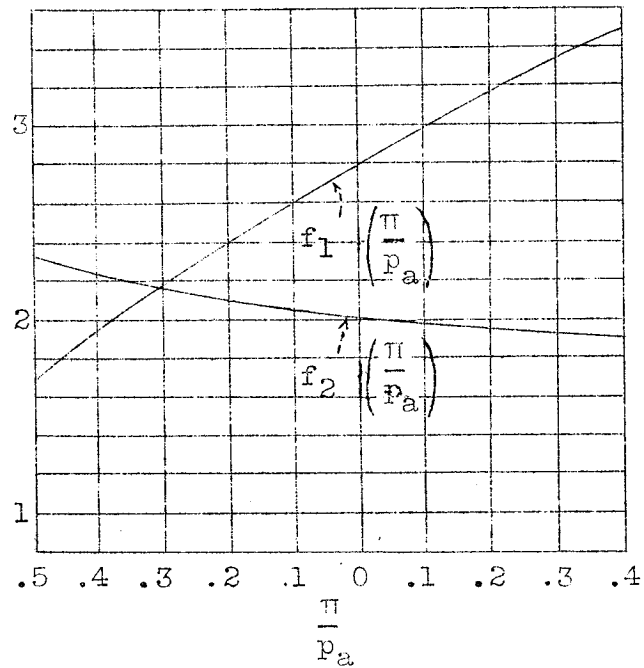


Fig. 11 $f_1 \left(\frac{\pi}{p_a} \right)$ and $f_2 \left(\frac{\pi}{p_a} \right)$ for $m_r = 1.4$

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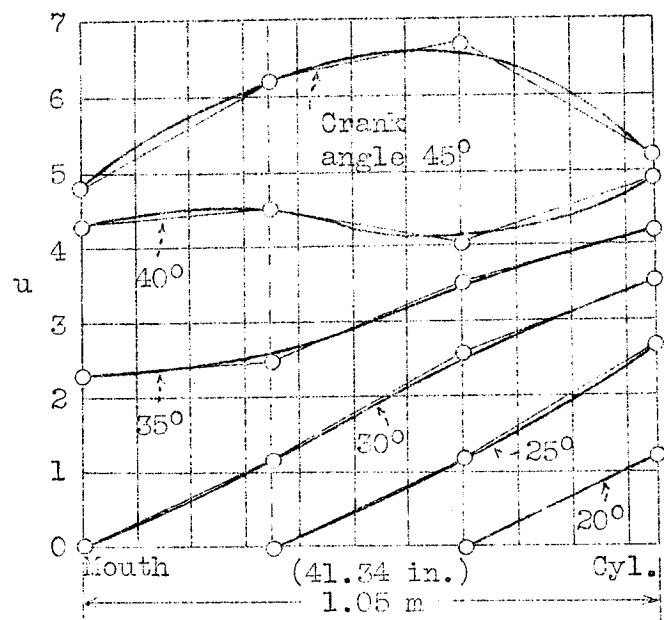


Fig. 12 Velocity variation along intake pipe of four-stroke engine.

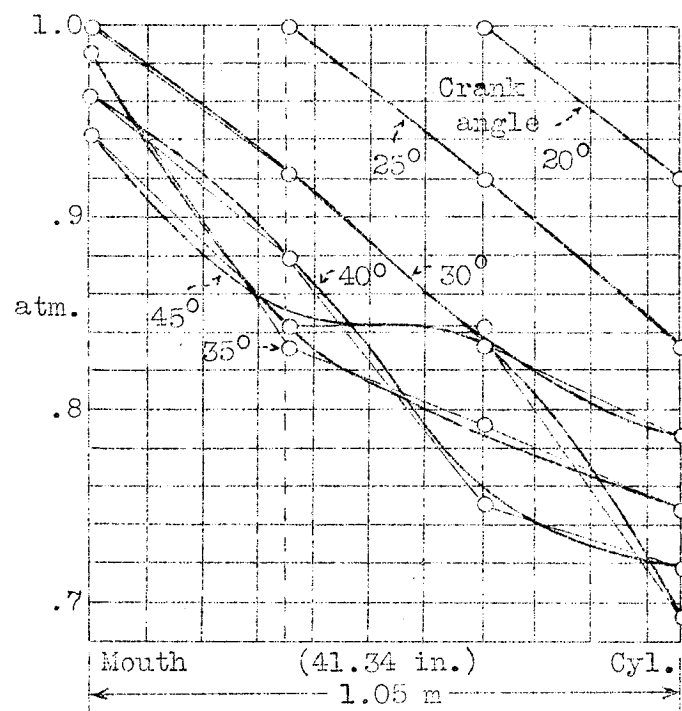


Fig. 13 Pressure variation along intake pipe of four-stroke engine.

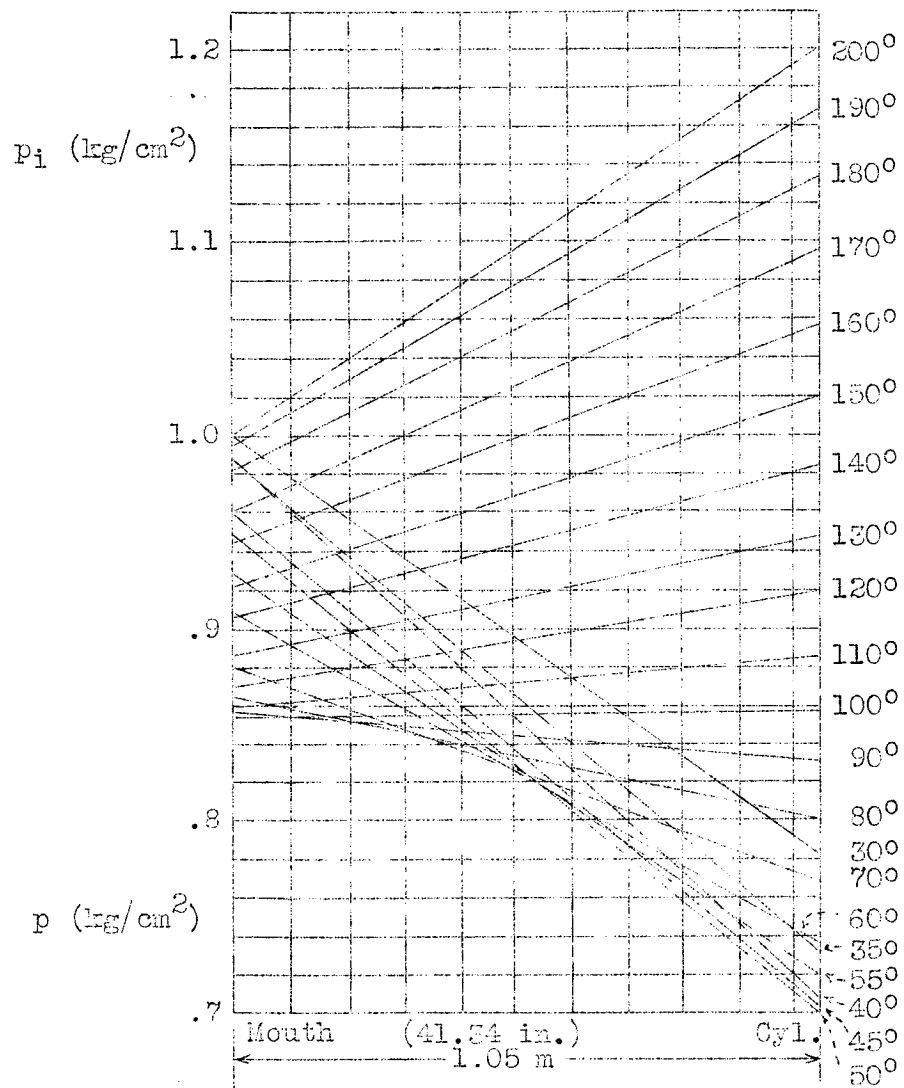


Fig. 14 Pressure during intake. Four-stroke engine.

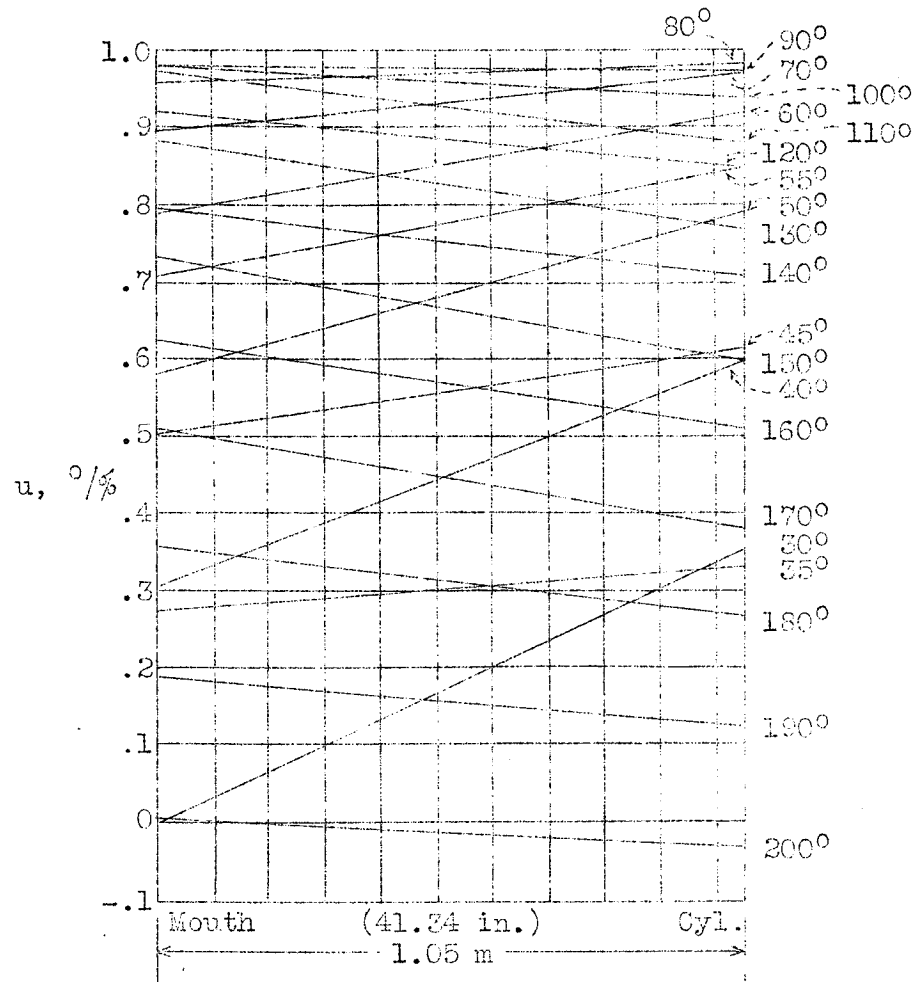


Fig. 15 Velocities during intake. Four-stroke engine.

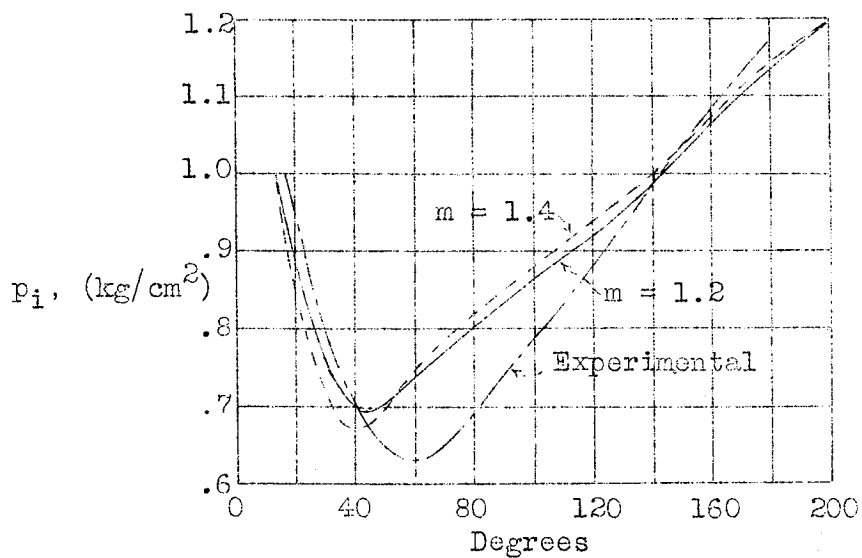


Fig. 16 Pressure in cylinder of four-stroke engine with intake pipe, plotted against crank angle.

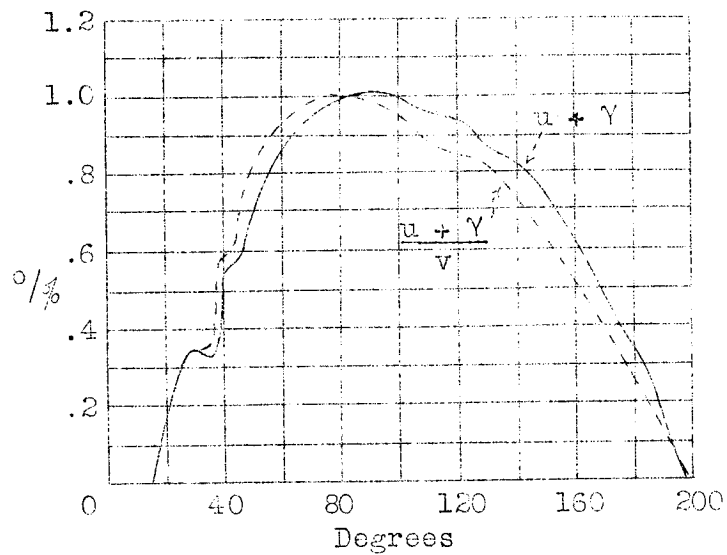


Fig. 17 Velocity at cylinder end of intake pipe of four-stroke engine, plotted against crank angle.

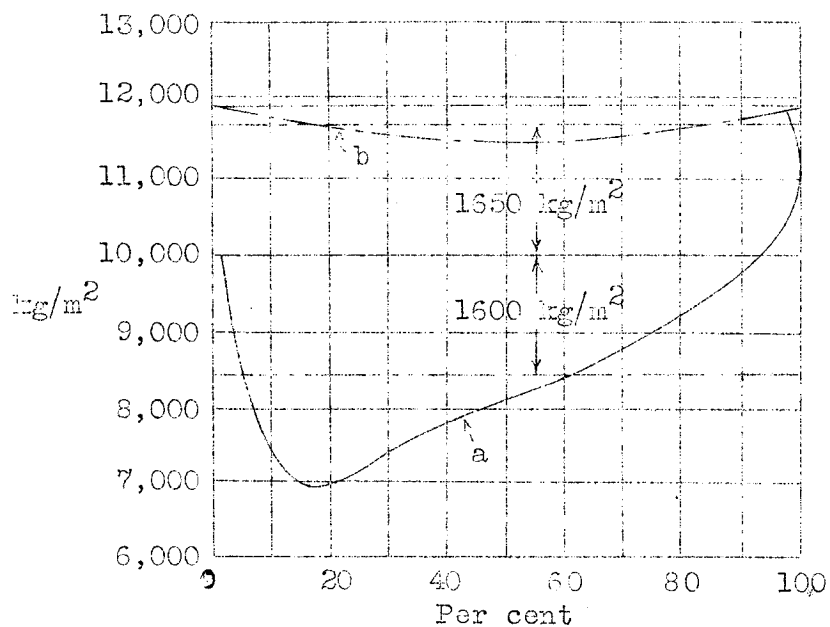


Fig. 18 Intake curve. a with intake pipe; b with supercharger. Four-stroke engine. (Piston travel)

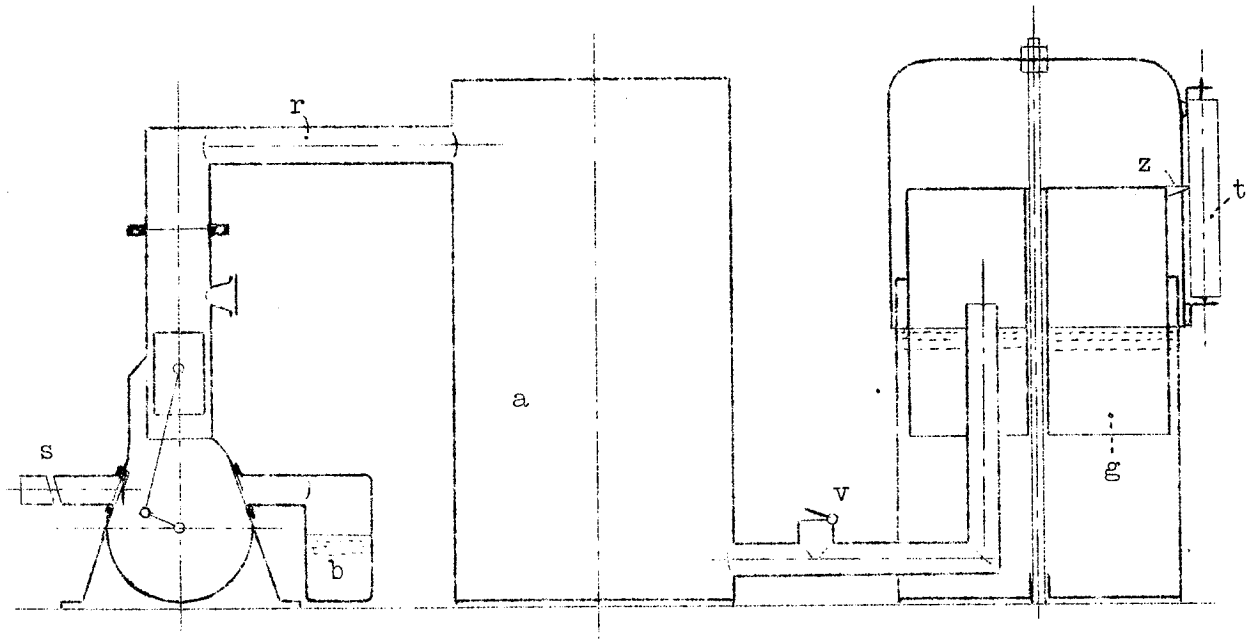


Fig. 19

Test installation

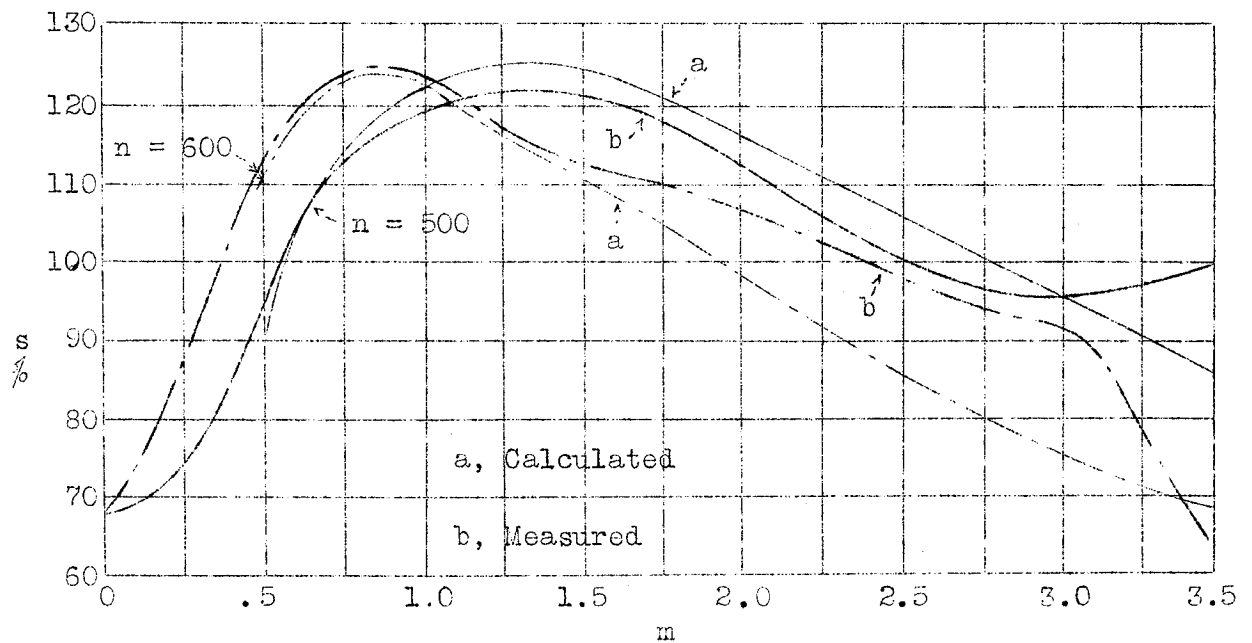


Fig. 20 Volumetric efficiency with controlled intake orifice. Diameter of intake pipe 1-3/4 inches

1. The first part of the report
describes the general situation
of the country and the
main problems which
are facing it.

2. The second part of the report
describes the results of the
survey which was carried out
in the different parts of the
country. It shows that the
situation is very serious
and that the people are
suffering from poverty and
ill health.

3. The third part of the report
describes the measures which
have been taken to improve
the situation. It shows that
the government is doing its
best to help the people and
that the situation is slowly
improving.

4. The fourth part of the report
describes the future plans
for the country. It shows
that the government is
determined to continue to
improve the situation and
to help the people to
live a better life.

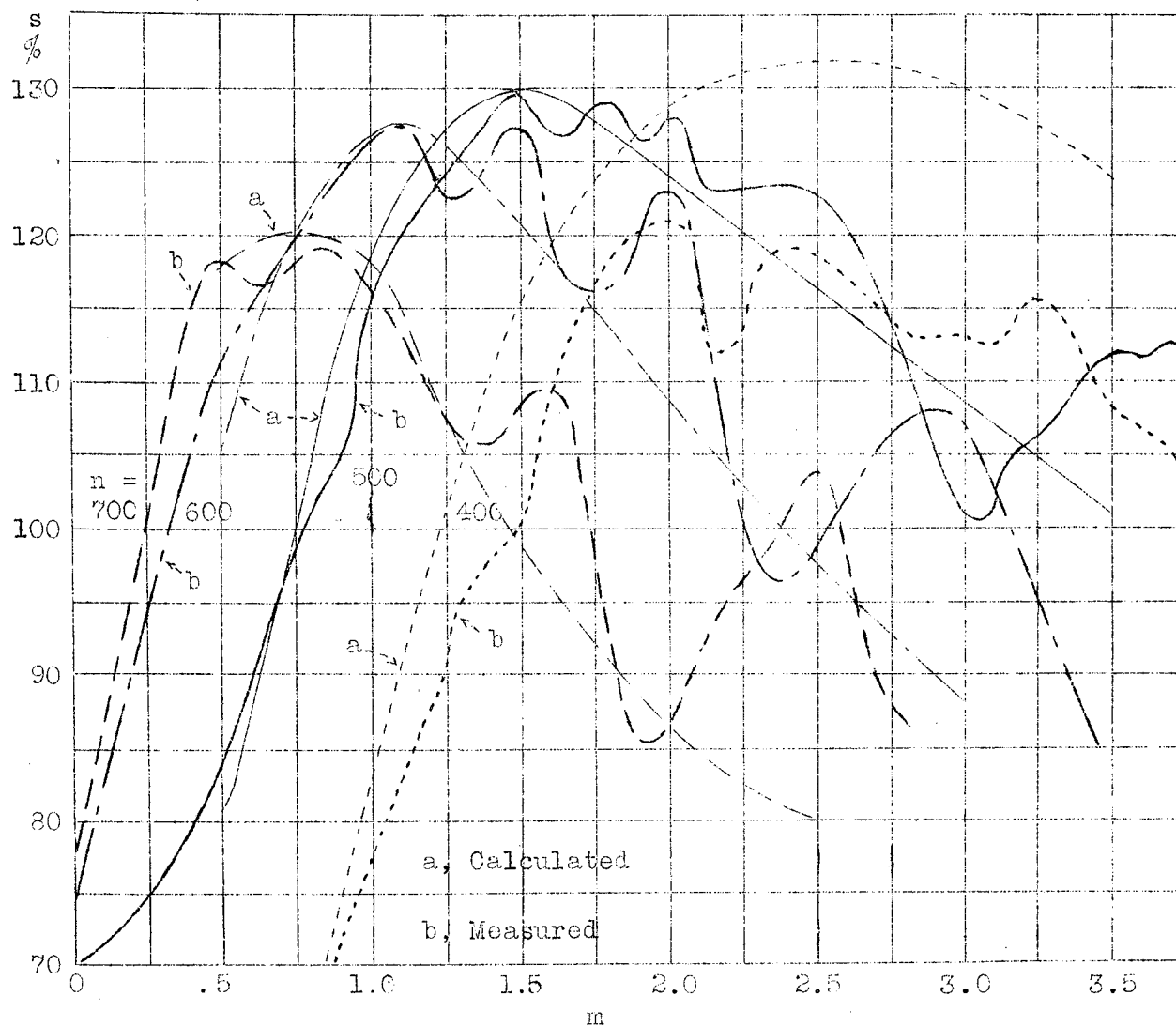


Fig. 21 Volumetric efficiency with controlled intake orifice. Diameter of intake pipe 2 inches.

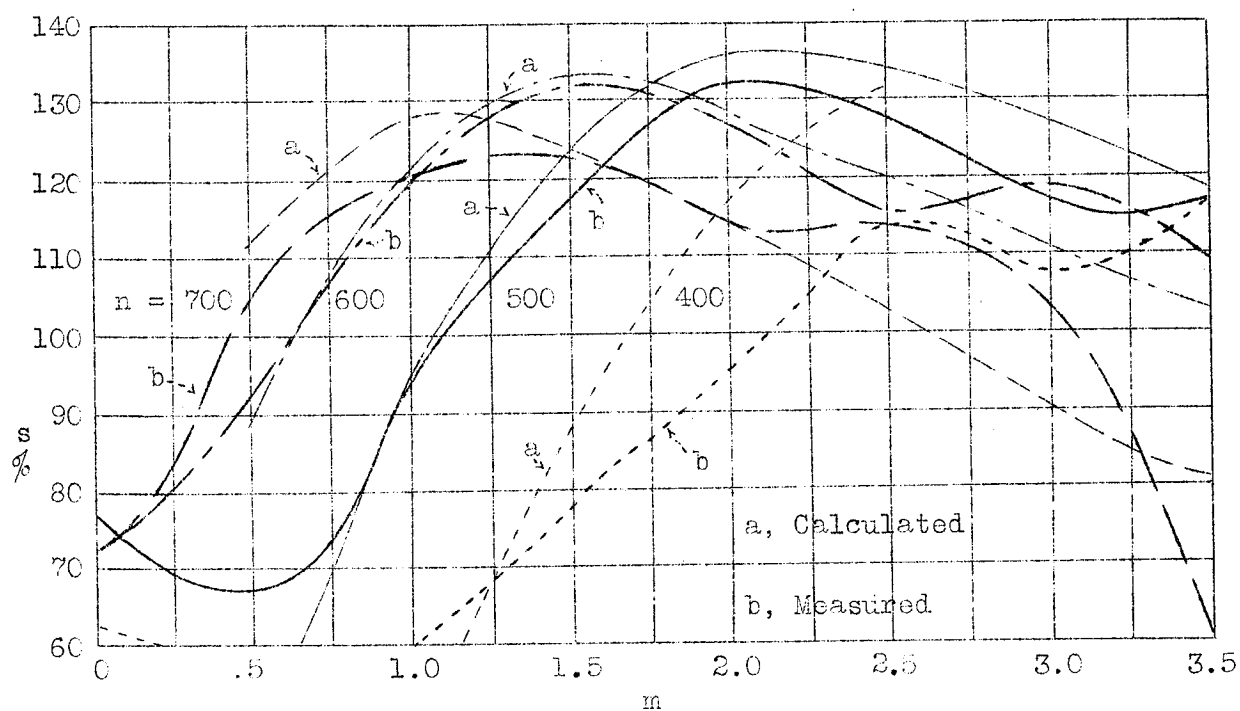


Fig. 22 Volumetric efficiency with controlled intake orifice. Diameter of intake pipe 2-5/8 inches.

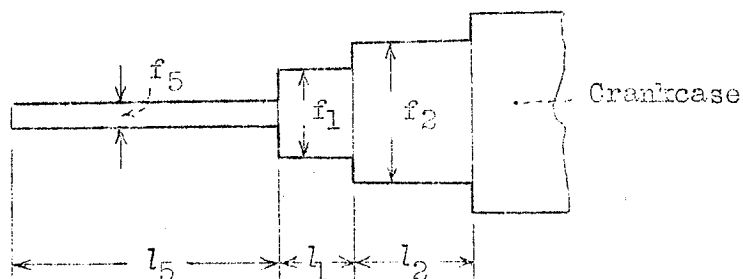


Fig. 23 Diagram of intake system.

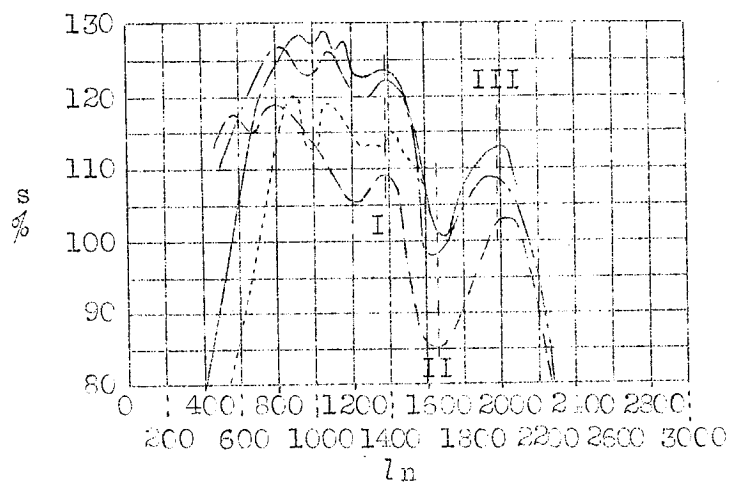


Fig. 24 Volumetric efficiency plotted against l_n . Controlled intake orifice. Diameter of intake pipe 2 inches.

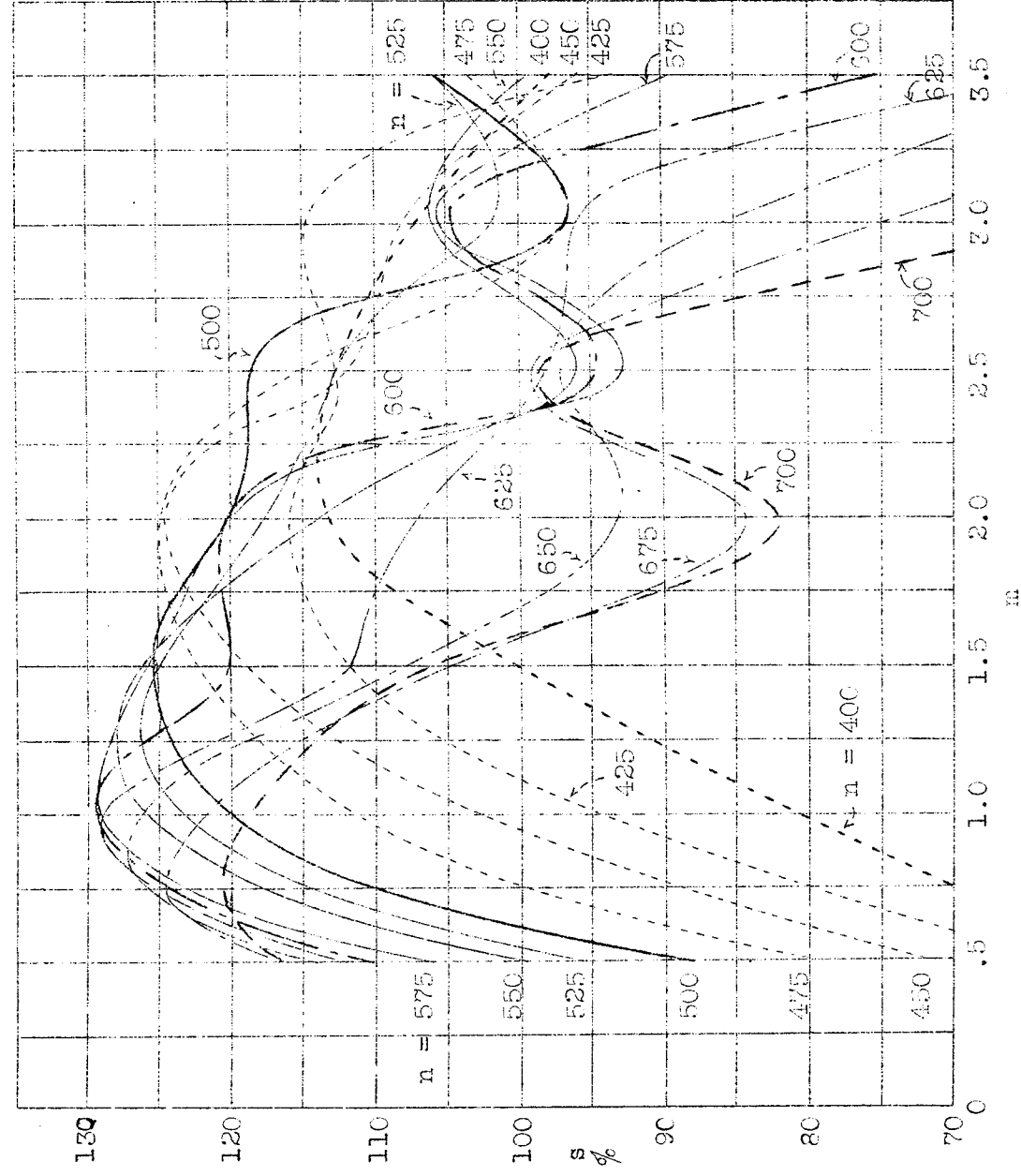


Fig. 25 Volumetric efficiency with controlled intake orifice. 2 inch intake pipe. Small revolution-speed gradations.

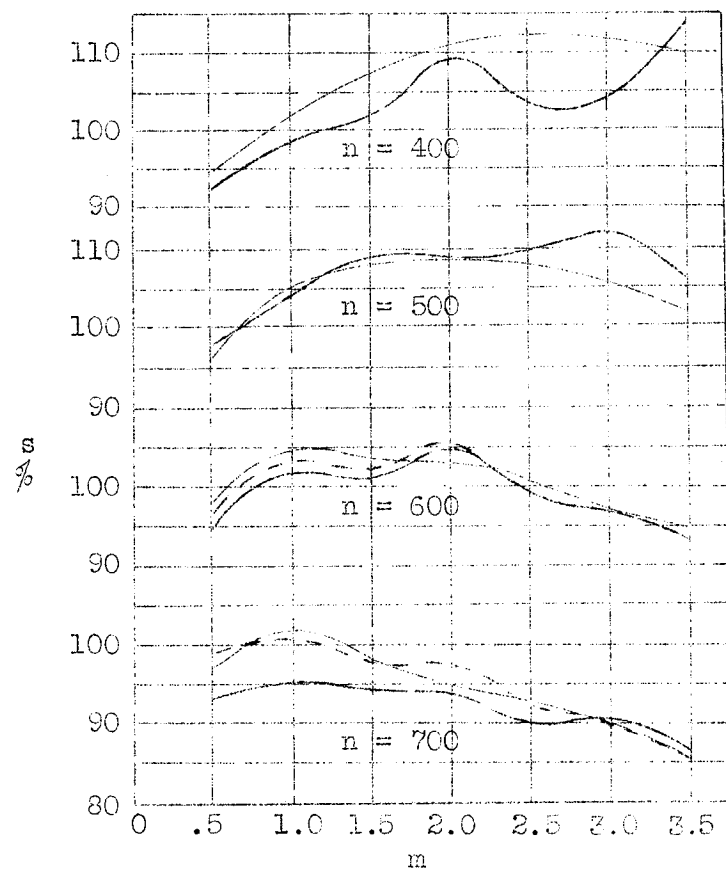


Fig. 26 Volumetric efficiency with automatic intake valve. 1 1/2 inches intake pipe. $x = 500\%$

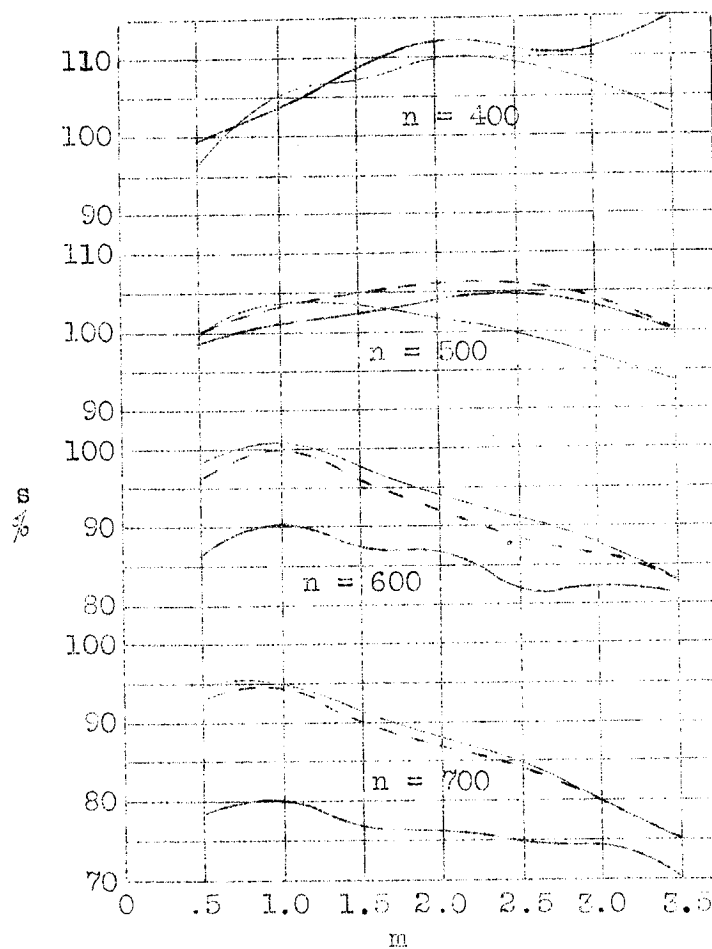


Fig. 27 Volumetric efficiency with automatic intake valve. 1 1/2 inches intake pipe. $x = 800\%$

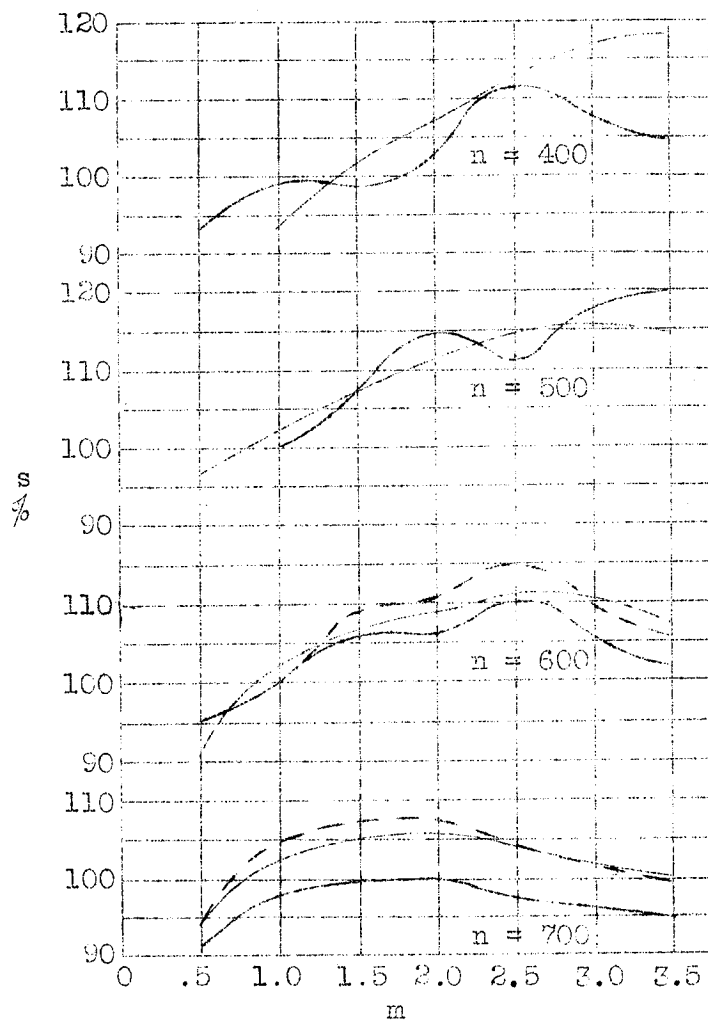


Fig. 28 Volumetric efficiency with automatic intake valve. 2 inch intake pipe.
 $x = 500\%$

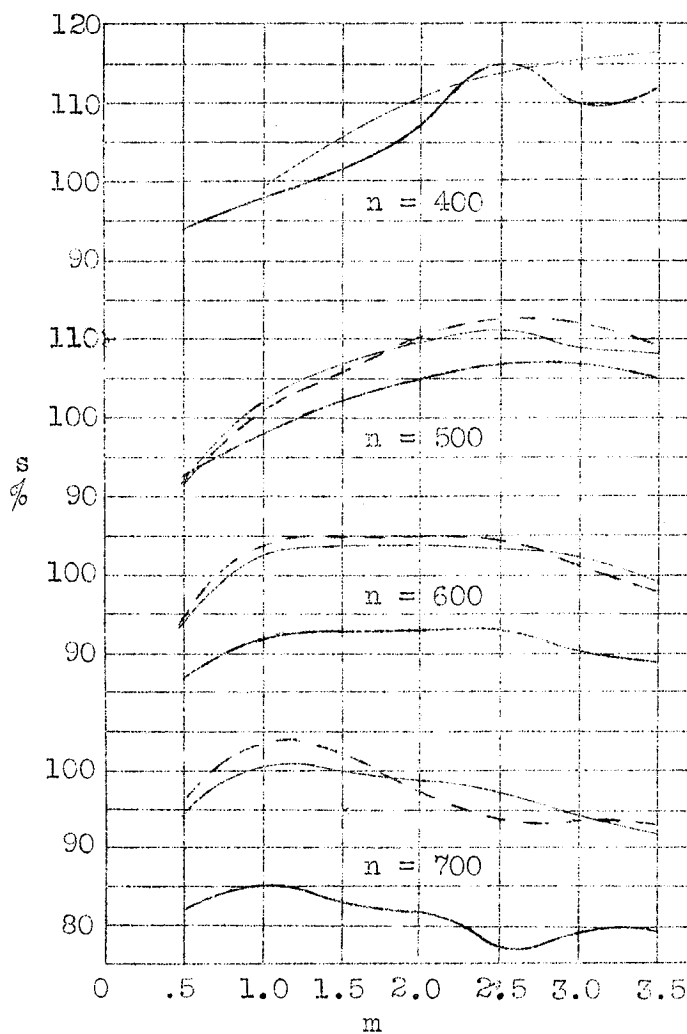


Fig. 29 Volumetric efficiency with automatic intake valve. 2 inch intake pipe
 $x = 800\%$

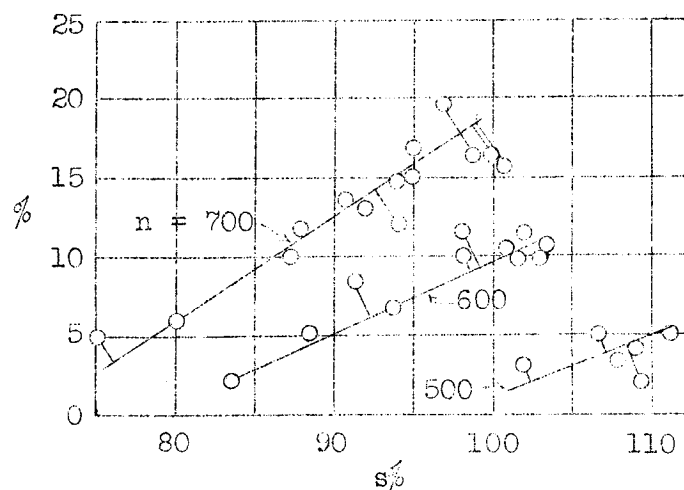


Fig. 30 Experimental errors in terms of revolution speed and calculated vol. eff. $x = 800\%$

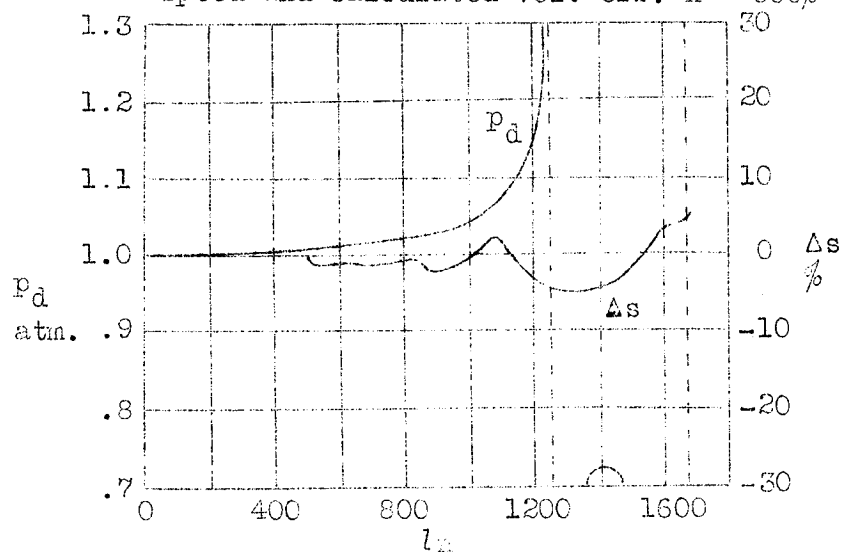


Fig. 31 Deviations of measured from calculated data due to vibration II. Final pressures with undamped vibration II.

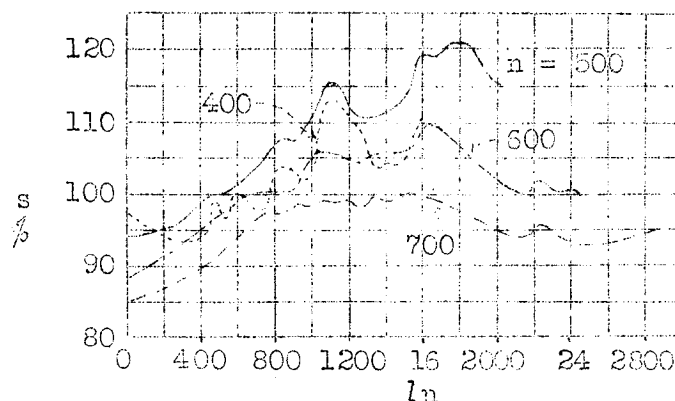


Fig. 32 Volumetric efficiency plotted against l_n . Intake valve. 2 inch intake pipe.

